COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

JANUARY, 1950

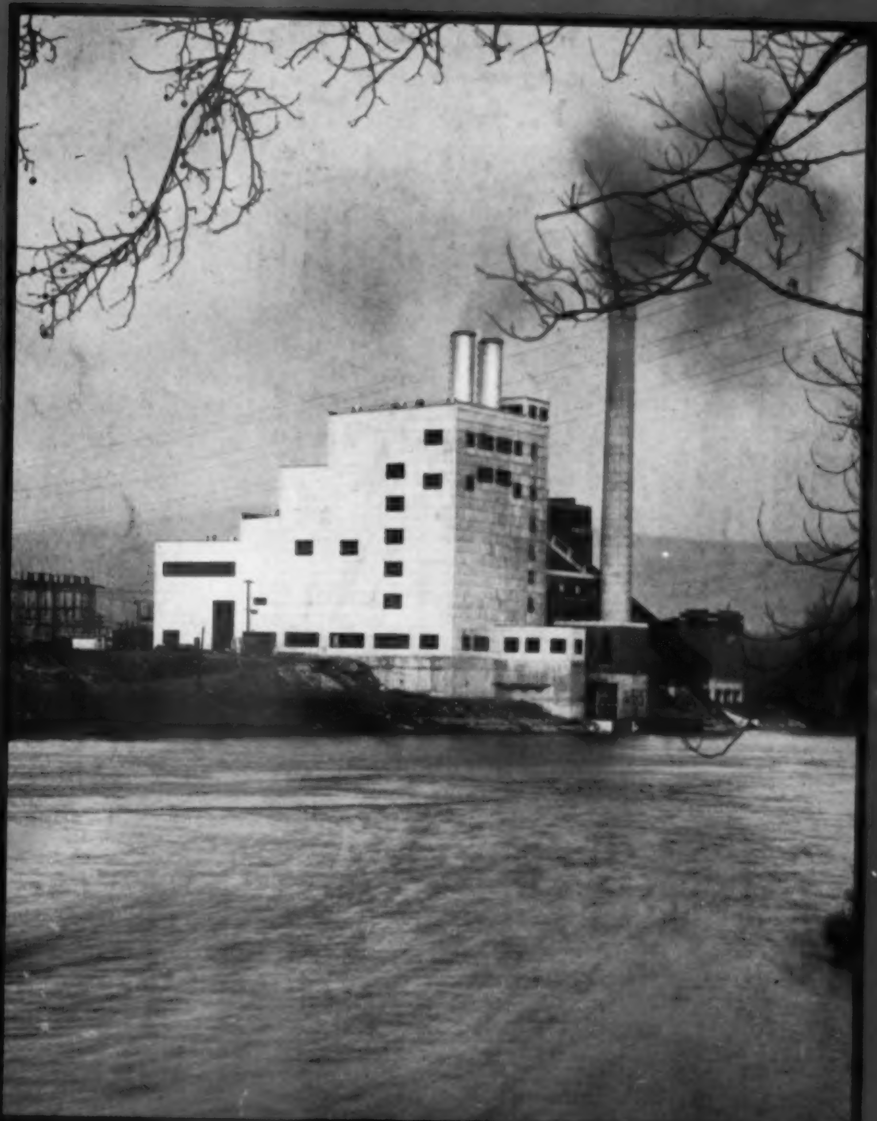

Photo by Patric P. Jacoskis

Gilbert Station of New Jersey Power and Light Co.

Gilbert Cross-Connects Conventional and Reheat Boilers

1949 Port Washington Experiences

COMBUSTION

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COMBUSTION

Editorials

Utility Production

Except for a brief period last fall, weekly peak demands for electric energy in 1949 were well above those of the corresponding weeks of 1948; and the total output of approximately 291 billion kilowatt-hours, as estimated by the Edison Electric Institute, was some three per cent above that of the preceding year, despite an apparent mild recession in some industrial lines.

The utility industry is now in a much better position to meet increasing demands than it was a year ago, for during 1949 nearly seven million kilowatts of generating capacity was installed and the average reserve has been more than doubled to a present figure of nearly twelve per cent. Furthermore, some five million kilowatts of additional capacity is scheduled for installation during the current year.

It is significant that 1949 marked a substantial shift from coal burning to oil and gas as fuel for utility boilers, coal having dropped from 54 per cent to 46.5 per cent. The permanency of this shift is likely to depend to a large extent upon the future actions of Mr. Lewis.

Still another point worth noting at this time is the fact that total fuel costs appear to have been somewhat less, despite increased output, thus reflecting the increased efficiency of the new stations, with their large units and higher steam conditions, that went into service during the year. This should be still more apparent during the present year with the commissioning of other new stations, particularly those employing the reheat cycle.

From the production standpoint, the utility outlook would appear most promising.

Another Year at Port Washington

In its January issue, for some years past, COMBUSTION has been privileged to report the preceding year's performance of Port Washington Station of the Wisconsin Electric Power Company.

This station, when designed nearly twenty years ago, incorporated a number of features far in advance of contemporary practice. Chief among these was employment of a single large boiler per turbine and very low furnace heat release rates. It also employed the reheat cycle in conjunction with the unit arrangement of boiler and turbine, and steam conditions of 1390 psi, 825 F. That these, together with several other features, all of a conservative nature, were sound is attested by the outstanding performance achieved by the station during its fourteen years of service, for it was not placed in regular service until near the end of 1935, due to deferred construction during the early depression years. In view of this performance, each succeeding extension has practically duplicated the first, except that the fifth unit, now building, will employ somewhat higher pressure and steam temperature.

It will be noted from the tables that the overall plant performance during 1949 differed little from that of the preceding year, although the fourth unit which has been in service four months showed a net heat rate under ten thousand Btu per kilowatt-hour. This should be bettered by the fifth unit because of the higher steam conditions.

With a use factor of nearly 95 per cent during 1949, Port Washington showed a demand availability of 95.6 for boilers and 95.2 for turbines. Forced outage has been low.

It is a tribute to those responsible for the basic design that this station is able to hold its front rank place with the latest post-war stations. In fact, it is doubtful whether any coal-burning steam station in this country or abroad has yet equaled its record on the basis of overall complete station performance for a full year.

The Creative Realm of Engineering

A few years ago the late A. R. Stevenson, Jr., inspired a series of discussions before the A.S.M.E. on the importance of ingenuity, intuition and creative ability in the engineering profession. These were reprinted in July 1944 under the title "Creative Engineering" and together with an article by William H. Easton entitled "Creative Thinking and How to Develop It" provide a valuable reference for a phase of engineering that has received less attention than it deserves.

E. G. Bailey has rendered a distinct service in renewing interest along similar lines through the James Clayton Lecture entitled "Inventing and Sifting Out Engineering Facts," before the Institution of Mechanical Engineers in London, excerpts from which are included in this issue. As one who has made notable contributions to the steam power field, he is particularly qualified to analyze the incentives and techniques of the inventive process.

In any consideration of invention and creativity two factors that constantly come to mind are the importance of imagination and intuition and the necessity for hard work. Logical thinking has its limitations which can be surmounted by men of vision who have the ability to combine intuition and reason to create innovations and inventions. But this process is not a simple one for it requires almost unceasing mental effort. In the words of Mr. Bailey, "Invention and a 40-hour work week are strangers."

Engineering is an art as well as a science. There is danger that the ever-increasing complexity of technology may obscure the human qualities which have contributed to past technical advances. More attention should be given to the encouragement and development of ingenuity and creativity, both during formal college training and in the early years of engineering employment.

Gilbert Extension Cross-Connects Conventional Steam Generator With Two Reheat Boilers

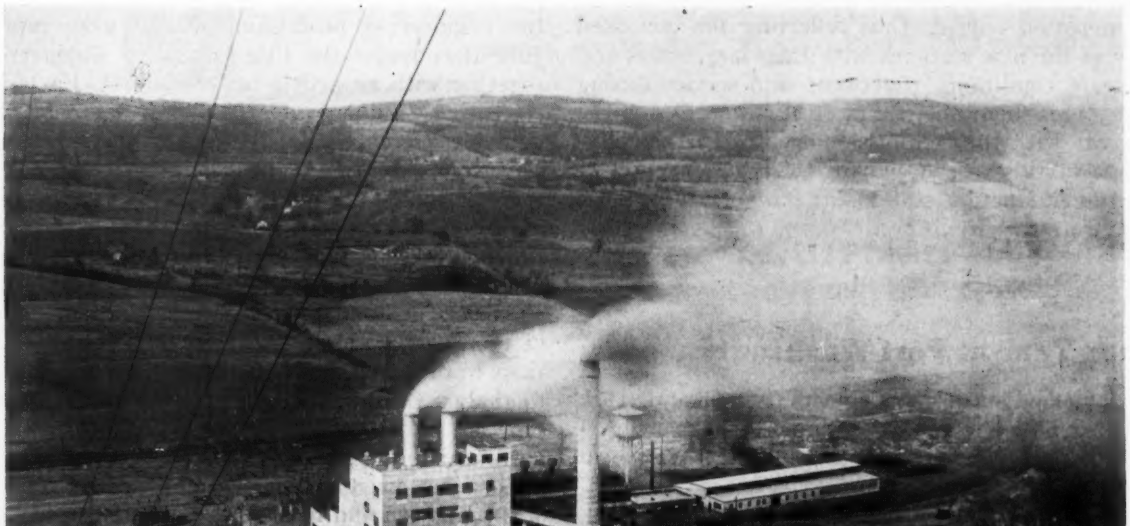
By T. Y. Mullen, Sponsor Engineer
Gilbert Associates, Inc.

The extension includes a 60,000-kw Preferred Standard turbine-generator supplied with steam at 1250 psig, 950 F by a 650,000-lb-per-hr pulverized-coal-fired steam generating unit of conventional design. This is cross-connected, through a desuperheating station, with two existing 1250-psig, 750-F reheat boilers serving a 55,000-kw turbine-generator. How this was accomplished is described. The overall cost of the plant extension, including substation equipment, was approximately $170 per kw.

DESIGN of the new extension to the Gilbert Station of the New Jersey Power and Light Company was started in 1941. The original plans called for the installation of two 200,000-lb per hr, 850-psig, 875-F boilers and one 35,000/43,750-kw straight condensing turbine-generator with no cross-connections between the existing plant and the proposed extension. Orders were placed for the major pieces of equipment on this basis.

Outbreak of the Second World War resulted in discontinuance of this project when a priority for materials was not granted by the War Production Board which then had jurisdiction over such matters.

This project was reopened at the close of the war and new studies were made by Gilbert Associates, Inc., to determine the relative merits of a single boiler versus a two-boiler installation; the advisability of burning anthracite on traveling grate stokers, or pulverized an-



Gilbert Station showing extension with tall stack

thracite versus pulverized bituminous coal; and the relative merits of higher pressures and temperatures as opposed to 850 psig, 875 F total steam temperature, which were the original design conditions.

These studies indicated that availabilities of conservatively designed modern steam generators very closely approximate those of turbine-generators, and that overall economy could be achieved by a single-boiler, single-turbine installation. They also showed that, although stoker-fired anthracite would result in the lowest operating and installation cost, the supply of low-cost stoker sizes of anthracite would be exhausted in this area within approximately seven years of the date of initial operation of the new extension. It would therefore be necessary to design these boilers to make them readily convertible to spreader-stoker firing to permit the use of bituminous coal at the end of that period. Estimated costs for conversion of these units to spreader-stoker firing, plus allowances for loss of generating capacity during the period of change-over, showed that these costs would approximately "break even" with the estimated savings of the seven-year anthracite-firing period.

Pulverized Anthracite vs. Bituminous Coal

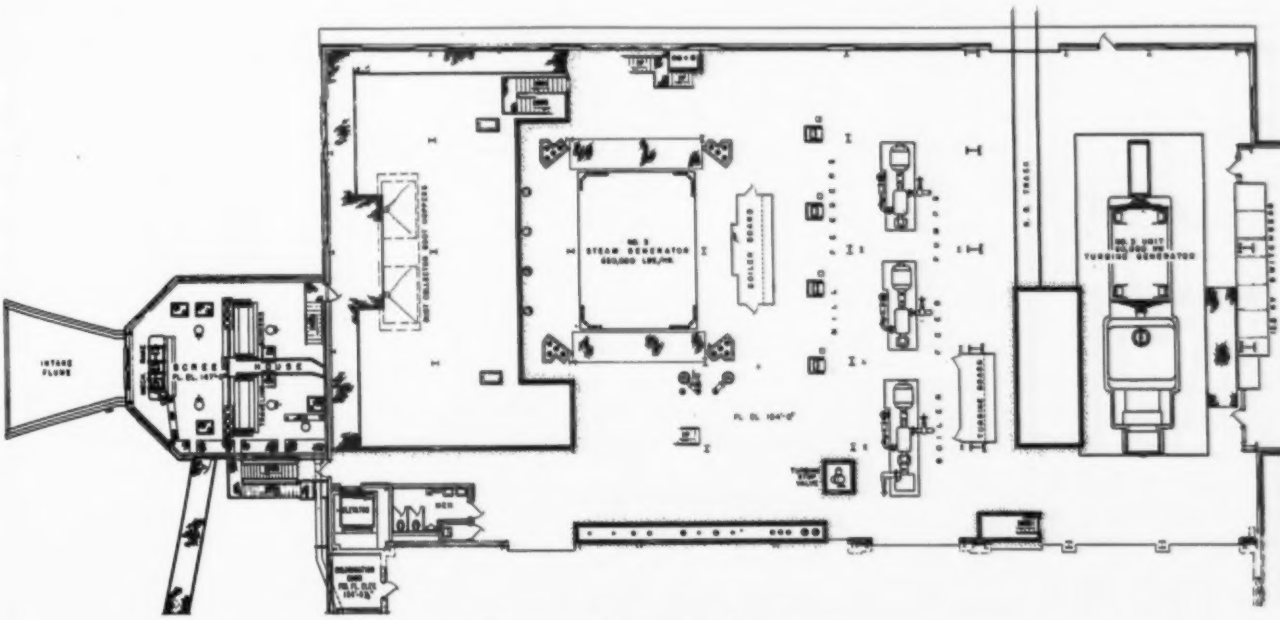
Pulverized anthracite did not appear economical at this plant due to the increase in furnace volume and consequent building volume required, the higher initial cost of major boiler-room equipment, the increased power consumption of the pulverizers, the lower boiler availability of units whose records were obtainable, the difficulties to be encountered in the handling and storing of the small anthracite sizes and maintaining two separate fuel storage piles, and the limited area and number of sources from which a reliable, long-term supply of low-cost anthracite fuel could be obtained. Therefore, pulverized bituminous coal was selected.

The rapidly increasing cost of all available fuels delivered at the Gilbert Station resulted in the decision to fix the initial steam conditions at 1250 psig and 950 F total steam temperature.

Units in Original Plant

The existing boilers at this station were two 250,000-lb per hr, 1250-psig, 750-F units with reheat to 750 F, operating in parallel to serve one 55,000-kw cross-compound reheat turbine-generator. These furnaces are primarily water-cooled but contain sections of refractory settings. Serious difficulties had been encountered during the war years and immediately thereafter, due to the accumulation of slag on these refractory sections as a result of the poorer grades of coal which it was necessary to burn at that time.

Studies of costs involved in changing of these furnaces to complete water-cooling, and the contingent cost of loss of generating capacity during the extensive change-over periods, indicated that these revisions were not economical at this time. It was, therefore, decided to increase the capacity of the new steam generator by 100,000-lb per hr and to cross-connect the new and existing main steam headers through a non-return valve and a desuperheating station. The old boilers would then be normally operated at loads of 200,000-lb per hr, which rating they had previously proved capable of maintaining with poor coal, without excessive slagging. The additional 100,000 lb of steam per hour required for the 55,000-kw turbine-generator was to be supplied by the new steam generator.



Plan at turbine room elevation

The new steam generator was ordered as a 460,000-lb per hr, 1250-psig, 950-F unit. A 40,000/44,000-kw, 1250-psig, 950-F Preferred Standard turbine-generator and a 100,000-lb per hr, 1250-psig, 950/750-F desuperheating station were ordered to be served by this steam generator.

Load growth on the New Jersey Power and Light System and that of other companies in the neighboring territories was considerably ahead of normal anticipated

growth; hence the quickest way to increase the installed capacity in this area was to increase the size of the unit on order for Gilbert Station. Checks with manufacturers of the major equipment disclosed that the shop work had not progressed to such an extent as to prevent making this change without delaying shipment. Therefore, the steam generator capacity was increased to 650,000-lb per hr, and the turbine-generator was changed to a 60,000/66,000-kw, 1250-psig, 950-F Preferred Standard unit. The desuperheating station remained unchanged. This is the equipment that was finally installed in the new plant extension.

The decision to cross-connect a straight condensing cycle, with a reheat cycle comprising two reheat boilers serving a single turbine-generator was reached only after careful consideration of the design and operating problems involved.

Problems of Cross-Connection

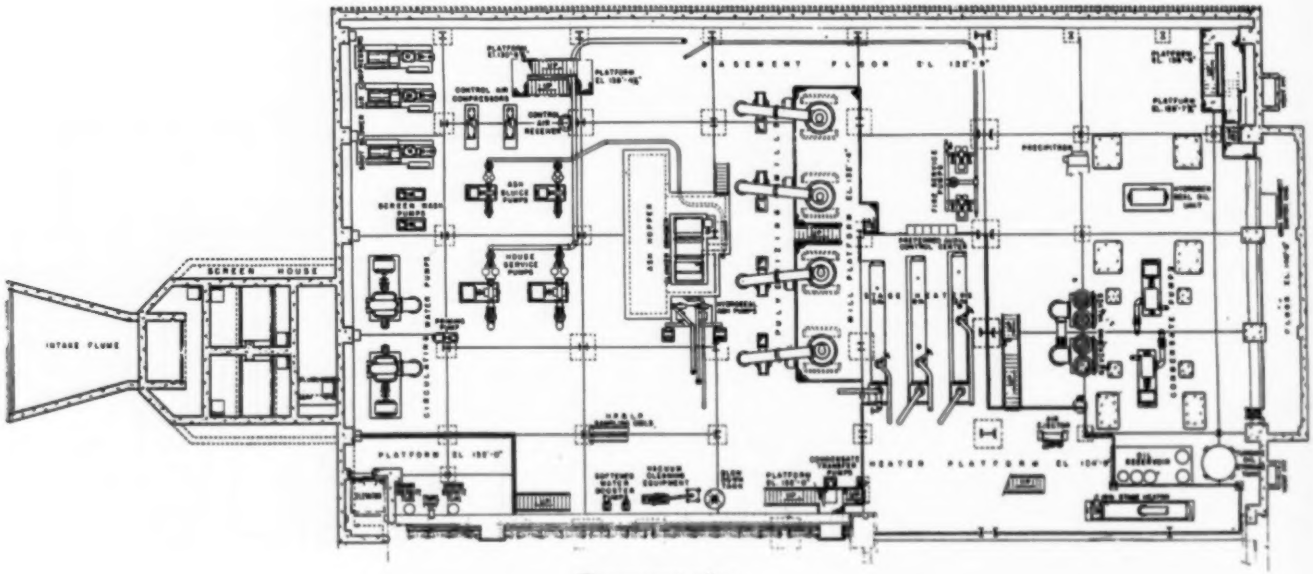
One of the most generally accepted practices for designing new reheat turbine-generator installations is that of adhering strictly to a single-boiler, single-turbine arrangement. This is based primarily upon the operating difficulties involved in bringing the second reheat boiler on the line, without burning up the superheater or reheater and without wasting excessive steam through blowoffs and drains. It also results from the problem presented in balancing the flow from the exhaust of the high-pressure turbine cylinder in proportion to the respective firing rate of each boiler feeding steam to that turbine. The unit system design was not used on the original installation at the Gilbert Station, however, and a three-port flow control valve is installed to regulate the steam flow to the reheaters.

Each of the existing boilers is equipped with a steam reheater and a gas reheater arranged in series. The steam temperatures leaving the gas reheaters have always been sufficiently high as not to require the steam reheaters to do any work. This condition makes it possible to operate the reheat boilers at reduced ratings; to put 100,000 lb of steam per hour from the new steam generator, together with the superheated steam output from the old boilers, through the high-pressure cylinder of the reheat turbine; and then pass all of the throttle steam for the low-pressure turbine through the reheaters without effecting a reheater pressure drop any greater than normally encountered at full rated reheat boiler load. By utilizing the steam reheaters, the low-pressure turbine throttle temperature can be maintained at 750 F without upsetting the heat balance of the reheat boilers.

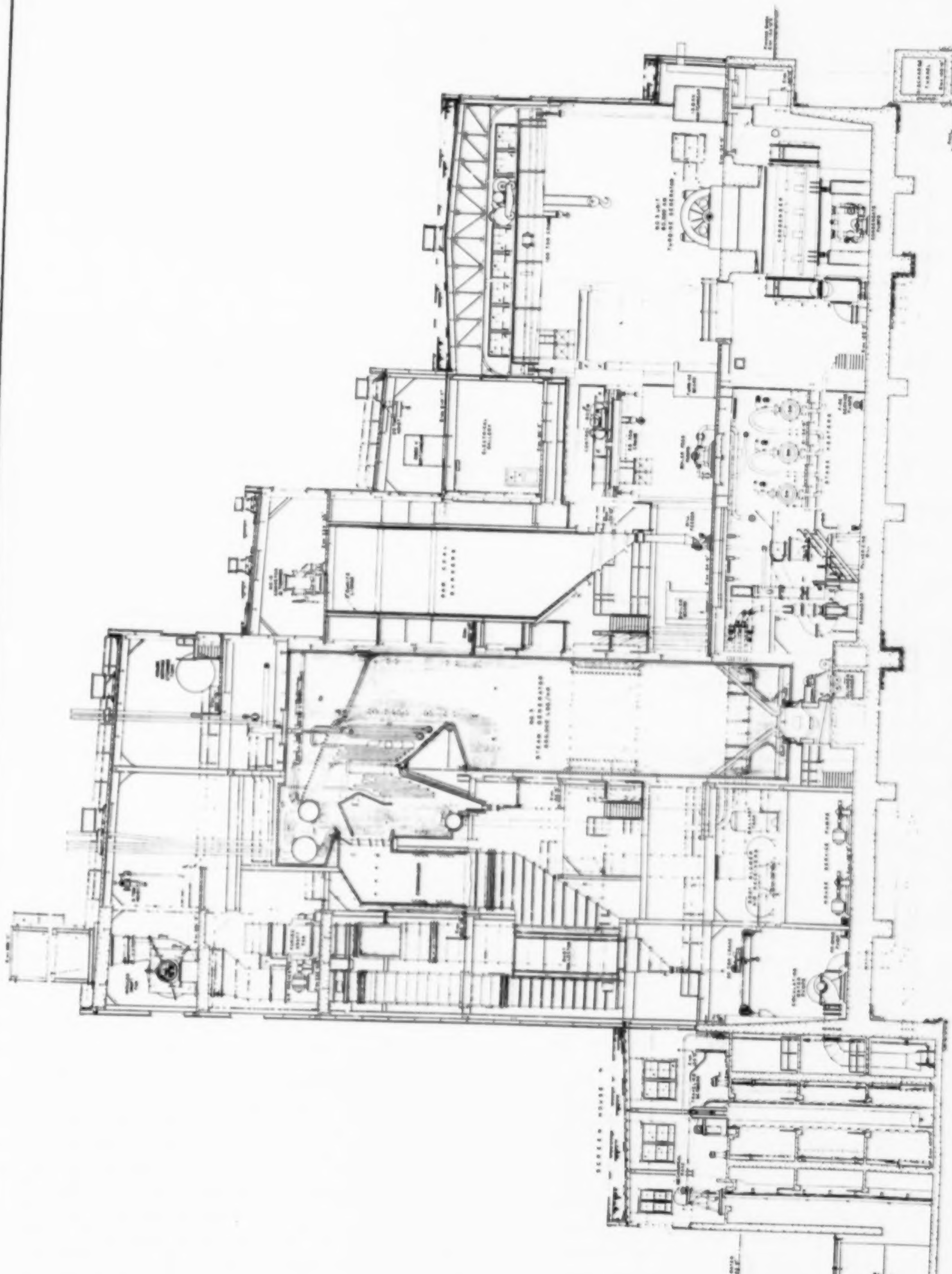
A bypass valve has been installed around the reheaters to permit bypassing some of the steam, when steam is being fed from the new boiler to the old high-pressure turbine, at a time when only one of the reheat boilers is in service.

The steam passes directly from the new main steam header through a desuperheating station to the old main steam header. The two turbines are designed for the same throttle pressure and the pressure drop through the desuperheating station must therefore be a minimum. This condition makes it necessary for the new boiler to operate in parallel with the reheat boilers. The combustion control equipment on both the new and the old units must have maximum sensitivity in order to keep the new steam generator from carrying all the load. The combustion control equipment for the existing boilers was, therefore, modernized to increase its speed of response. In addition, the slight operating pressure differential between the new and the existing steam header presented a problem of controlling the rate of flow of steam from the new station to the old one. A front setter was accordingly installed on the master controller for the new steam generator to permit the operator to vary his new header pressure as required to maintain the desired steam flow to the old steam header.

The condensate leaving the old condenser flows to the deaerating heater through a surge tank and a float valve



Basement plan



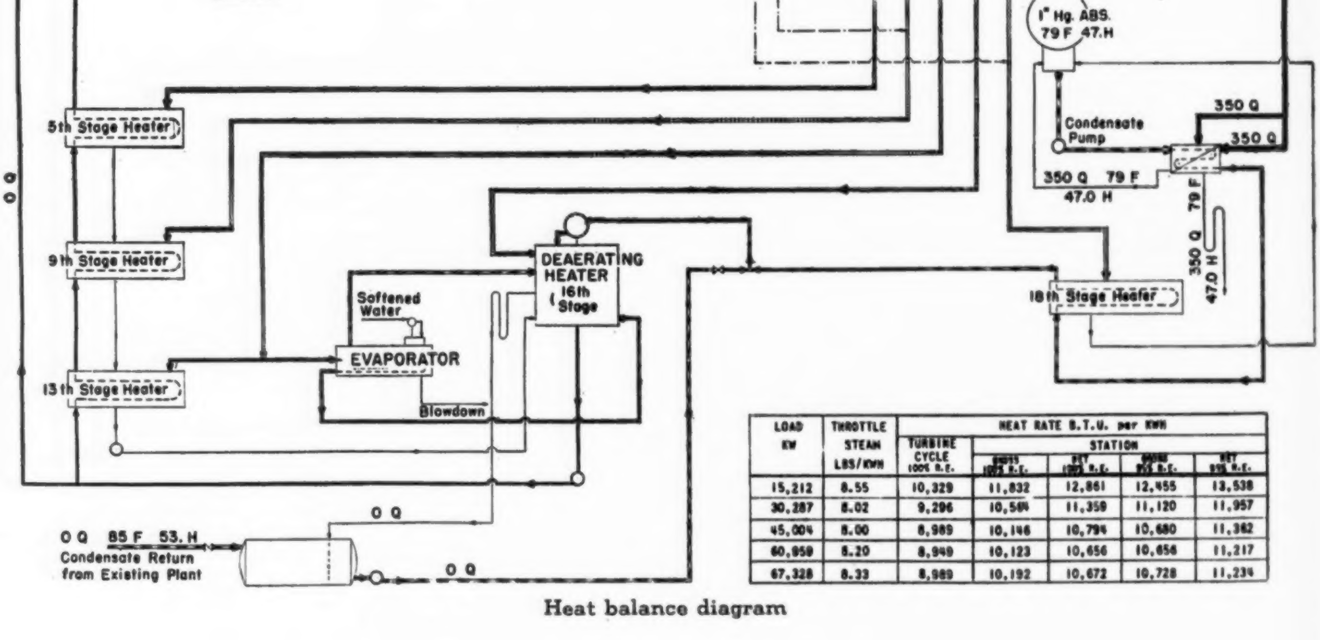
Sectional elevation through plant extension

with the excess flowing to the condensate storage tank. This flow to the storage tanks represents primarily the flow of steam from the new boiler to the old turbine, and is transferred by the condensate transfer pumps through a float valve to the new deaerating heater. The deaerating heater and the three high-pressure heaters for the new station are designed to heat 100,000 lb of condensate per hour from the old cycle in addition to all of the condensate for the new turbine cycle.

New Plant Equipment

The new plant contains one 650,000-lb per hr, 1250-psig, 950-F Combustion Engineering-Superheater tangentially-fired, pulverized-bituminous-coal-fired steam generator; a 60,000/66,000-kw, 1250-psig, 950-F Preferred Standard General Electric turbine-generator, served by a 50,000-sq. ft., twin, two-pass Ingersoll-Rand surface condenser equipped with divided water boxes and two MacNeill reversing valves; two 1000-gpm horizontal centrifugal condensate pumps; one low-pressure Foster Wheeler closed feedwater heater equipped with an integral drain cooler; one twin element, two-stage air ejector unit mounted on a separate surface inter and after condenser; one 650,000-lb per hr Elliott floating-type deaerating heater equipped with stainless steel trays; three 765-gpm Ingersoll-Rand boiler feed pumps, two motor-driven and one dual-driven by a motor on one end and through a Rawson spring-controlled spacer coupling on the other end from an 1100-hp, 3600-rpm Westinghouse impulse turbine supplied with steam at 1250 psig, 750 F and exhausting at pressures varying from 10 in. Hg abs. to 15 psig; three Foster Wheeler high-pressure closed feedwater heaters each equipped with an integral drain cooler; and one Copes, two-element, remote manual or automatic feedwater regulator.

The circulating water for the new extension is taken from the Delaware River and passes through a new screen house where it is screened and chlorinated before passing to two 24,250-gpm horizontal Ingersoll-Rand centrifugal circulating pumps. The screen house is equipped with a Link-Belt mechanical trash rake to remove large pieces of material before the water passes to two Chain Belt four-post screens.



LOAD KW	THROTTLE STEAM LBS/KWH	HEAT RATE B.T.U. per KWH				
		TURBINE CYCLE 100% B.E.	STATION			
			GROSS 100% B.E.	NET 100% B.E.	GROSS 85% B.E.	NET 85% B.E.
15,212	8.55	10,329	11,832	12,861	12,455	13,538
30,287	8.02	9,296	10,564	11,359	11,120	11,957
45,004	8.00	8,989	10,146	10,794	10,680	11,362
60,959	8.20	8,949	10,123	10,656	10,656	11,217
67,328	8.33	8,989	10,192	10,672	10,728	11,234

Heat balance diagram

Coal enters the plant property by railroad and is either stocked by use of a skip hoist and drag scraper, or it is delivered to two car hoppers in the track hopper house. A Robbins car shake-out is used to vibrate the cars when unloading, and two sets of Hauck thawing pits have been installed to assist in handling frozen coal. The two track hoppers feed separate inclined Link-Belt conveyor belts having a capacity of 150 tons per hour each. These two belts feed a 300-ton per hr Bradford breaker which reduces the size of the coal to $1^1/_4$ in. and under. The coal is discharged from the Bradford breaker to two 150-ton per hr inclined belts equipped with magnetic pulleys and weightometers. These two belts discharge to a 300-ton per hr belt equipped with a tripper serving the raw-coal bunkers in the old portion of the plant, or to an inclined

cross belt which feeds the horizontal belt with tripper over the 1400-ton capacity raw coal bunker serving the new steam generator. The tops of the raw-coal bunkers in both the new and the old sections of the plant have Link-Belt seals to minimize coal dust in the bunker rooms. The new raw coal bunker is fabricated of $^3/_8$-in. steel plate lined with 2 in. of gunite to within 6 ft of each of its four 18-in. square outlets. The bottom 6 ft of each bunker is fabricated of monel-clad steel plate to minimize coal arching, to reduce wear due to abrasion, and to prevent corrosion due to weak sulfuric acid formed by moisture and sulfur in the coal.

The bunker has been designed with one vertical side continuing down to the outlet of each of the four pockets to prevent arching and rat-holing and to make the bunker self-cleaning. The four bunker outlets discharge through four downspouts to four C-E floor-type feeders equipped with Speedtrol drives. These downspouts are 18 in. square at the top, have vertical centerlines and taper to 22 in. square at the bottom to facilitate coal flow and minimize sticking and packing in the spouts. The feeders discharge through vertical spouts to four C-E Raymond bowl mills, each having a nominal capacity of 16,450 lb per hr of 55 Hardgrove grindability, 10-per cent moisture content, coal ground to 80 per cent through 200-mesh screens. Each pulverizer is equipped with an exhauster which discharges to one burner in each of the four corners of the furnace.

Ash is collected in a flooded hopper below the furnace and sluiced to an ash pit immediately adjacent to the ash hopper. The ash passes through a clinker grinder and is then pumped to a settling area by either of two Allen-Sherman-Hoff "Hydroseal" ash pumps. Fly ash is collected in an Aerotec tubular-type dust collector. The dust is continuously removed from the two hoppers under the last pass of the steam generator and from the two hoppers below the boiler by four "Hydromix" valves and then flows by gravity to the ash-settling area.

Air for combustion is taken from the top of the boiler room or from the outside atmosphere by two Sturtevant forced-draft fans and delivered to two Ljungstrom regenerative-type air pre-heaters before passing to the pulverizers and the windboxes. Each forced-draft fan has a capacity of 100,000 cfm of 100-F air at a static pressure of 12.6 in. of water, is direct connected to a 350-hp, 1200-rpm motor and is inlet-vane controlled.

Hot gases are removed from the furnace and economizer through the dust collector and the two air pre-heaters and delivered to two 40-ft gunite-lined steel stacks by two Sturtevant induced-draft fans, each equipped with a double-extended shaft and direct-connected to an 800-hp, 900-rpm motor on one end and to a 250-hp, 600-rpm motor on the other end. Each induced-draft fan has a capacity of 192,000 cfm of 315 F gas at a static pressure of 16.9 in. of water at 880 rpm and a capacity of 127,000 cfm of 273 F gas at a static pressure of 7.4 in. of water at 585 rpm.

Makeup water for the entire station is pumped from the Delaware River to settling basins before entering gravity-type filters. The filtered water to be used for makeup is then pumped through zeolite softeners to two evaporators. One evaporator serves the original turbine cycle and another evaporator serves the new cycle. Both discharge their vapor to their respective deaerating heaters. The cross-connecting of the two turbine cycles makes it possible for either evaporator to serve both units. The new Lummus evaporator is equipped with an integral deaerating heater to pre-heat and deaerate the softened water entering the evaporator, and with a reflux condenser and bubble tray to wash the leaving vapor.

The power plant extension group includes an extension to the main building, housing one 60,000-kw turbine-generator, one 650,000-lb per hr steam generator, and all auxiliary equipment; an extension to the turbine operating floor for a 13,800-volt switchgear room, and a new screen house to serve the new unit and cross-connected with the screen house serving the old unit.

The design of all buildings includes reinforced concrete foundations, structural steel framing and asbestos siding. The original plant is brick.

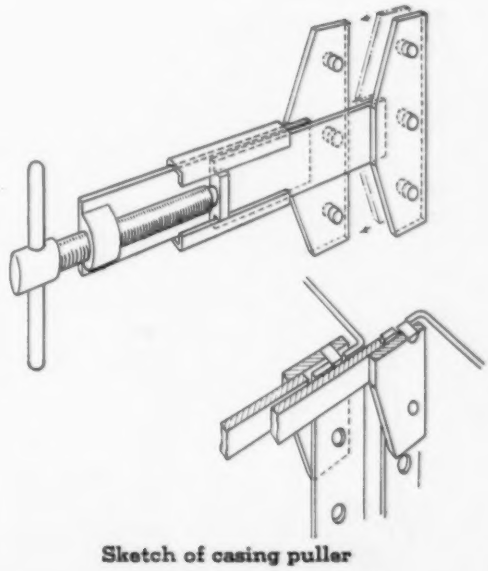
The overall cost of the plant extension including substation equipment was approximately \$10,600,000, or about \$176 per kilowatt of new capacity added. It is estimated that approximately \$200,000 was saved through the use of asbestos siding in place of brick walls.

Gilbert Associates, Inc., Reading, Pa., were the engineers for the project and L. H. Focht & Son, Inc., were the general contractors. The new turbine-generator was placed in commercial operation on November 1, 1949.

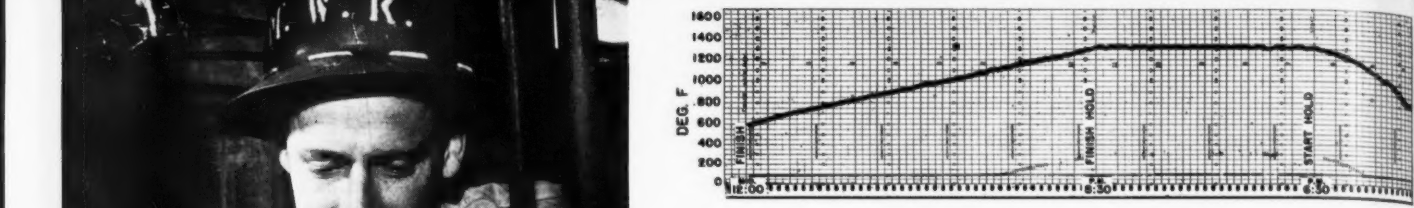
Boiler Casing Panel Puller

Boiler casings may be removed quickly and easily without the damage usually associated with that operation by means of a panel puller devised by Earl Bentley of the Service and Erection Department of Combustion Engineering-Superheater, Inc. The tool is simple in construction, easy to operate, and extremely rugged, and it enables panel removal with a minimum of defacement to any gaskets and brickwork that may be involved.

Components of the tool include two similar arms, a sliding clip, and a hand-operated screw. Operation of the hand screw causes the arms to move relative to one another, thereby effecting panel removal. The sliding clip permits disassembly of the tool into two parts, making possible application of tool to the flanged panel being removed. On each of the arms there are pins which match bolt-hole spacing and provide means of attaching the arms to the panel flange.



Sketch of casing puller



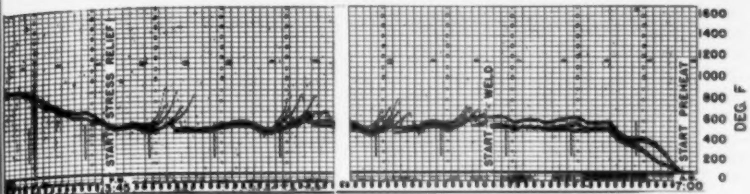
Field Welding H.-P.,

THIS sequence of eight photos shows the actual steps in welding and stress-relieving an alloy-steel pipe to a cast-steel gate valve in the South Meadow Station of Hartford Electric Light Co. Welding specialists of M. W. Kellogg Co., Inc., are seen field erecting the chrome-molybdenum piping for the main boiler-turbine steam header on Unit No. 6, which operates under throttle conditions of 850 psig, 900 F. The procedure is representative of the welding and heat-treating sequence used in numerous installations, and currently applicable to alloy piping for steam conditions as high as 2000 psig and 1050 F.

Fig. 1 shows the final lining up and spacing of a section of alloy pipe of A.S.T.M. designation A-280 and a cast-steel valve. The backing ring at the bottom of the groove is seamless and machined to very close limits. In the shop it is attached to the inside of the pipe by means of welding in a close-tolerance, machined counterbore. The welding inspector is using a standard spacer plug to check the space at the bottom of the groove.

The band appearing on the left of Fig. 1 is made of carbon steel and is welded to the outside of the pipe. It is used to identify the pipe, to designate such inspections as may be made, and to record the welders who make the weld at this point. An additional function is for attaching a protecting head on the end of the pipe for shipment. Elimination of stenciling and tack welding on the pipe by use of this protective band prevents the formation of localized surface stress concentrations.

Attachment of thermocouples and binding of leads is shown in Fig. 2. These are used to determine the temperature of the joint during the pre-heating, concurrent heating and stress-relieving operations. The asbestos blankets protect the thermocouple leads from the heat produced by induction heating.



H.-T. Alloy-Steel Piping

Fig. 3 shows welders installing additional asbestos blankets to protect the electrical cables which are to be wrapped around the pipe for induction heating.

As indicated in Fig. 4, 500,000 cir-mil insulated cables are wrapped around the pipe for induction pre-heating and stress relieving. Here two separate 40-kva transformers furnish the power.

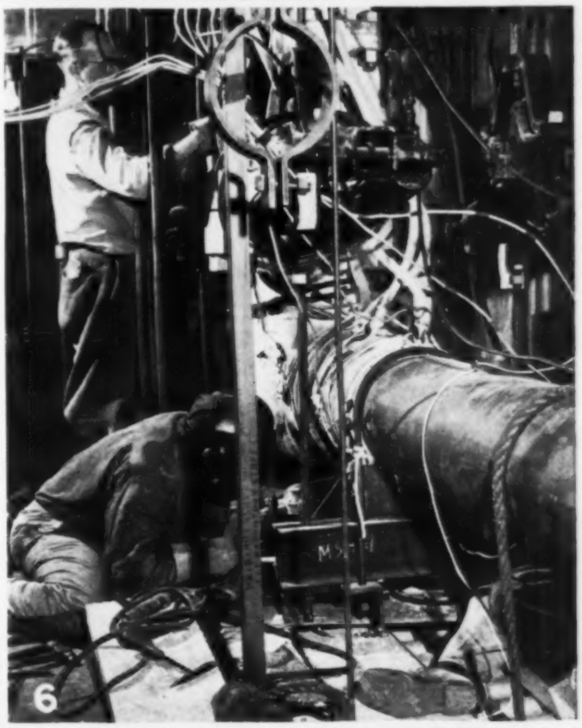
Nearly eight hours are required to make this weld, although the time element varies with pipe thickness. Fig. 5 shows the first bead being welded in by the two welders working on both sides of the joint. The welding inspector is closely watching the laying in of this bead which is possibly the most important part of the weld. Complete fusion of the ends of the pipe and valve must be made and penetration into the backing ring obtained.

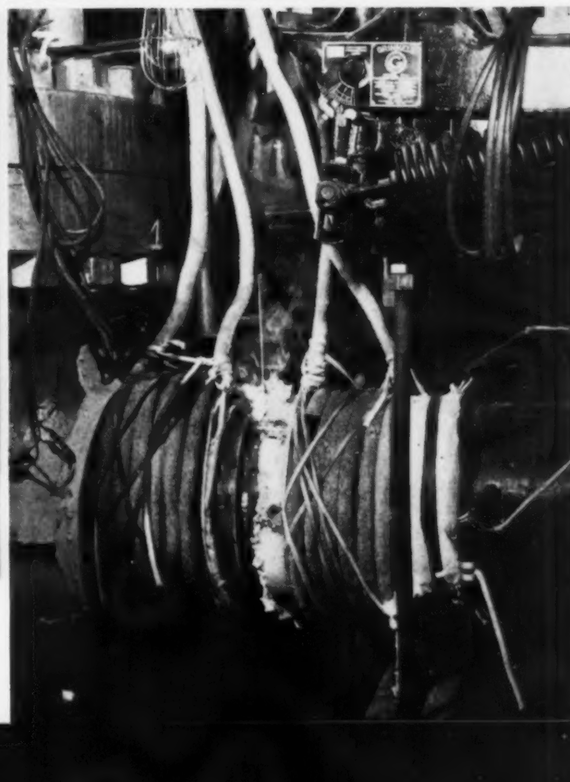
Succeeding weld layers are applied as shown in Fig. 6. The welders work from the bottom of the pipe to the top.

The slag deposit on the top of the bead being laid in may be seen in Fig. 7.

Fig. 8 shows the chipper cleaning the slag off this layer and preparing the surface of the weld metal for the laying in of the next bead.

The two curves at the top of the page are reproductions, reading from *right to left*, of sections from a pyrometer chart which recorded temperatures during the welding process, i.e. pre-heat, concurrent heat and the post-heat or stress-relieving stages. Five thermocouples are mounted on the pipe and a sixth serves as a reference base. The heating rate is 400 deg F per hr, and when a temperature of 1300 F is reached the temperature is held constant for two hours, after which the weld is allowed to cool to room conditions at a rate of 200 deg F per hr. The chart shows the uniformity of heating and cooling as well as the maintenance of constant temperature over extended periods, the result of extensive experience.





Photos by Dave Henderson through courtesy of The M. W. Kellogg Company, Inc.

Possibilities of The Regenerative Steam Cycle at Temperatures up to 1600 F

In this paper, which was presented at the recent A.S.M.E. Annual Meeting, the authors calculate the gains for both the theoretical cycle and for the practical cycle wherein such losses as extraction-piping pressure drop, heater terminal temperature differences, etc., are considered. An economic evaluation of the anticipated turbine heat-rate differences for various throttle conditions is presented, and comparison is made of heat-rate gains due to higher steam temperatures with those possible through resuperheating.

By P. H. KNOWLTON* and R. W. HARTWELL†

THE use of steam for electric power generation has been characterized by a fairly steady increase in steam temperature, which has made possible steady gains in thermal efficiency. The increase in maximum steam temperatures in power plants constructed during the last 45 years is shown in Fig. 1. This shows that the temperature has increased an average of 12 deg per year, but during the last 20 years temperatures have increased faster than this average.

Theoretical Possibilities at Higher Temperatures

Selvey and Knowlton[1] showed theoretical regenerative steam-cycle heat rates for temperatures up to 1200 F and pressures up to 3200 psia. To facilitate a comparison over a wide range of pressure-temperature conditions, these data have been extended herein to 1600 F over the higher range of pressures, using the same theoretical cycle. As stated in the Selvey and Knowlton paper, the heat rates are for a regenerative-steam cycle of the following component parts:

1. A turbine, having no mechanical losses, through which steam is expanded at constant entropy (engine efficiency is 100 per cent).
2. A regenerative feedwater system having an infinite number of bled-steam heaters heating to the saturation temperature corresponding to the throttle-steam pressure, and with zero terminal difference between the saturation temperature of the bled steam and the temperature of the feedwater leaving the heater, even when superheat is present in bled steam, and an attendant feed pump of 100 per cent efficiency with each heater to step up the feedwater pressure to the level of the next heater.
3. An electric generator of 100 per cent efficiency, which supplies, without line loss, the boiler feed pumps.
4. A steam generator of 100 per cent efficiency, in which blowdown, soot-blowing losses, etc., are zero.

[1] "Theoretical Regenerative Steam-Cycle Heat Rates," by A. M. Selvey and P. H. Knowlton, A.S.M.E. *Trans.*, vol. 66, August 1944.

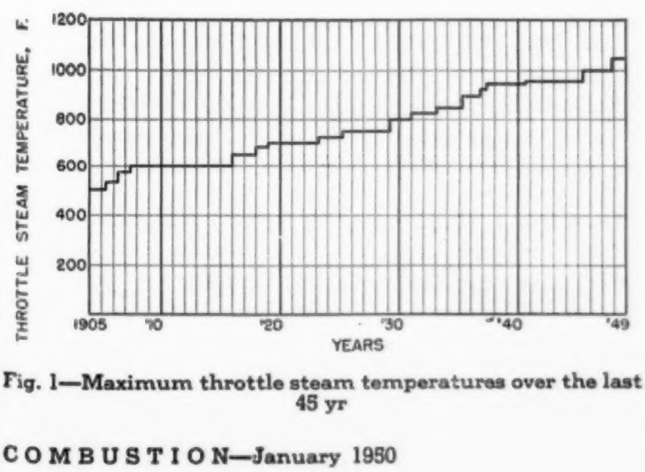


Fig. 1—Maximum throttle steam temperatures over the last 45 yr

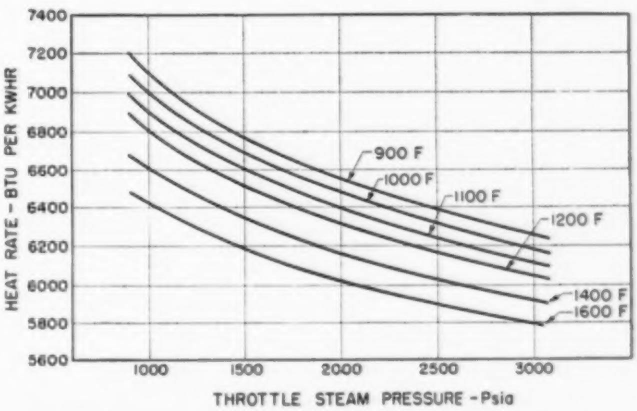


Fig. 2—Theoretical regenerative steam-cycle heat rates

Since energy required to drive fans, fuel equipment, general services, etc., is considered zero in a theoretical cycle, auxiliary-power usage other than that required for the boiler feed pumps, as in Item (2), is not included.

The theoretical heat rates are presented by the curves in Fig. 2. This information shows that there is a steady

* Turbine Engineering Division, General Electric Company, Schenectady, N. Y.
† Mechanical Engineering Division, The Detroit Edison Company.

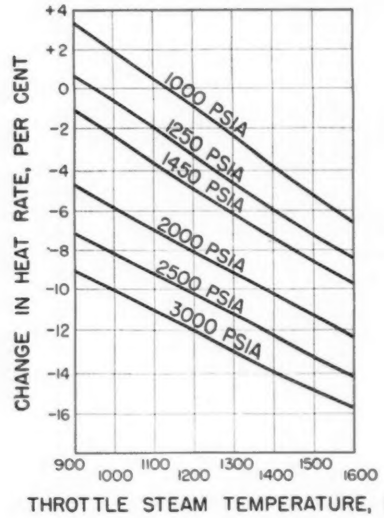


Fig. 3—Per cent change in theoretical-cycle heat rate vs. initial temperature. Base condition—1250 psia, 950 F and 1 in. Hg abs. exhaust pressure

improvement in the theoretical cycle economy with higher steam temperatures, but that the rate of improvement diminishes as the temperatures are increased. Fig. 3 presents the theoretical heat rate gains on a percentage basis with the 1250 psia, 950 F throttle condition as a base. These curves provide a means of comparing theoretical and practical heat rates.

Practical Possibilities at Higher Temperatures

Fig. 4 shows comparative per cent change in regenerative-cycle heat rates that are considered practical from the design and operating standpoints. These heat rate changes, computed for the same high-temperature and pressure range as was selected for the theoretical heat rate calculations, include the effects of such losses as extraction piping pressure drop, heater terminal temperature differences, etc. From a specified heat rate for a particular turbine generator, the heat rate for any other

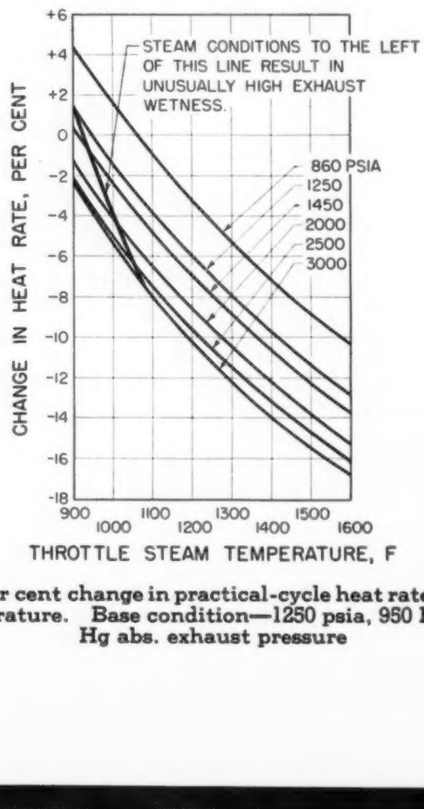
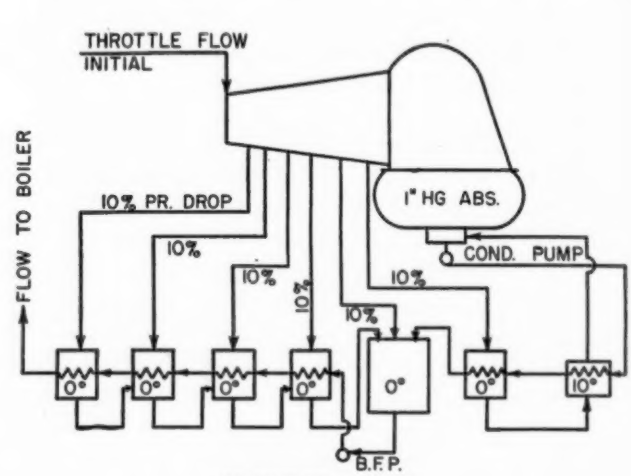


Fig. 4—Per cent change in practical-cycle heat rate vs. throttle temperature. Base condition—1250 psia, 950 F and 1 in. Hg abs. exhaust pressure

condition may be calculated with the data included in Fig. 4.

As in Fig. 3, the base steam conditions chosen are 1250 psia, 950 F. The per cent change in heat rate from this base condition to any other condition may therefore be read directly. If it is desired to obtain the correct difference in heat rate between any other two sets of steam conditions, in terms of the heat rate at one of these conditions, the difference as read from Fig. 4 must be corrected for the change in base. For example, if one wishes



ASSUMPTIONS

1- TERMINAL DIFFERENCES AS INDICATED

2- 10% PRESSURE DROPS FROM STAGE TO HEATER

3- STEAM PRESS.	860	1250	1450	2000	2500	3000
4- B.F.P. DISCH. PRESS.	1000	1450	1700	2300	2900	3450
5- ENTHALPY RISE IN B.F.P., BTU/LB.	4.1	6.3	7.6	10.3	12.9	15.4
6- NO. OF HEATERS (CHANGING NUMBER OF H.P. HEATERS)	4	5	6	7	9	10
7- BOILER FEED TEMP., F.	380	434	458	516	556	590

8- ENTHALPY RISE IN CONDENSATE PUMP 0.6 B.T.U./LB.

Fig. 5—Practical-cycle heater arrangement for heat rate calculations

to get the difference between heat rates at 1450 psia, 1250 F and at 1450 psia, 1000 F, in terms of the heat rate at 1450 psia, 1000 F, this difference is

$$\frac{7.9\% - 2.4\%}{1 - (2.4/100)} = \frac{5.5}{0.976} = 5.6\%$$

Practical Cycle Assumptions

In making the calculations to arrive at the values plotted in Fig. 4, the following assumptions have been made:

1. The cycle arrangement and associated apparatus have been assumed to be as shown in Fig. 5, which presents the same arrangement as was used by Harris and White.[2] The present calculations have been extended to higher temperatures and pressures than those assumed by Harris and White and are based on somewhat lower feedwater temperatures. Power required to drive the boiler feed pump has been deducted from the generator terminal output.

2. Turbine and generator efficiencies have been assumed to correspond to those obtained in present-day

[2] "Developments in Resuperheating in Steam Plants," by E. E. Harris and A. O. White, A.S.M.E. *Trans.*, vol. 71, No. 6, August 1949.

turbine-generators. The turbine-generator efficiency assumptions correspond to those made by Warren and Knowlton.[3] This implies that at temperatures above present-day levels, new features of turbine design, or new materials, or both, must be developed if the general level of efficiency obtained in the past is to be maintained.

3. The exhaust loss from the turbine has been assumed to be a constant fraction of the turbine power output for all steam conditions.

4. It is assumed that the turbine-generator output is about 100,000 kw. This must be borne in mind particularly when comparing the heat rates at various initial pressures, since turbines of smaller capacities than 100,000 kw probably would show poorer relative performance at the higher pressures.

Throttle and condenser steam rates, based on the practical cycle assumptions, have been computed to indicate the relative size of the generating equipment. The throttle and condenser steam rates are presented in

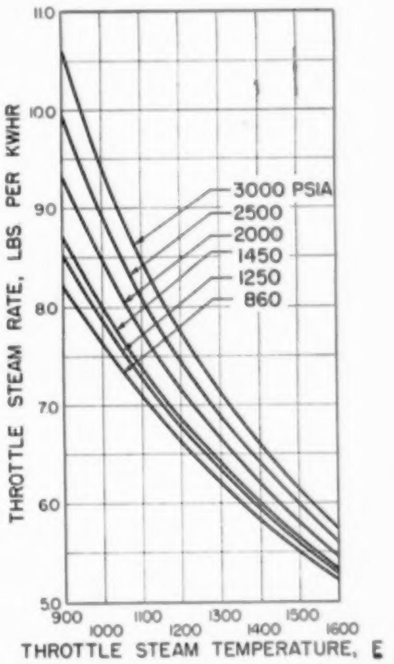


Fig. 6—Throttle steam rate vs. throttle steam temperature, 1 in. Hg abs. exhaust pressure

Figs. 6 and 7, respectively. These are shown on the basis of the generator terminal output, whereas the heat-rate change curves, Fig. 4, show the net change after deduction of boiler feed pump power from the generator terminal output.

Heat Rate Difference Evaluation

In considering the merits of one steam condition over another, it is desirable to evaluate the anticipated difference in heat rates. Because the practical heat rates represent results which at present appear possible at the higher steam conditions, these anticipated heat rates were used in this evaluaton rather than the theoretical heat rates.

The basic factors included in this evaluation are the cost of fuel, load characteristics, auxiliary power require-

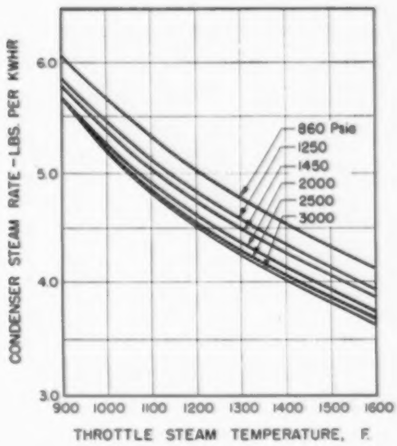


Fig. 7—Condenser steam rate vs. throttle temperature

ments, boiler efficiency, turbine heat rates and annual fixed charges on plant investment. The table presents the results of an evaluation of heat-rate differences wherein a base condition of 1250 psia, 950 F has been used for comparing all the other conditions. The figures shown represent the additional investment in dollars over the cost of 100,000 kw of 1250-psia, 950-F capacity that is justified by the differences in heat rates. In computing the data it was assumed that plant availability and operating and maintenance personnel would be the same in all cases. No attempt was made to allow for possible differences in the extent of maintenance required.

Cost of Fuel: A 10-cent to 35-cent per million Btu fuel cost range has been covered in the evaluation.

Load Characteristics: An annual plant factor variation from 50 to 80 per cent has been included, the annual plant factor being the ratio of the kilowatt-hours generated on the machine in a year to the product of the kilowatt nameplate rating times 8760.

Auxiliary Power Requirements: The following types of pulverized-coal-fired boilers were assumed in computing fan and mill power requirements.

(*a*) For throttle pressures up to an including 2000 psia —natural circulation boilers.

(*b*) For 2500 psia throttle pressure—"Steamotive" cycle boilers with recirculating pumps which handle 130 per cent of steam flow.

(*c*) For 3000 psia throttle pressure—once-through forced-circulation boilers.

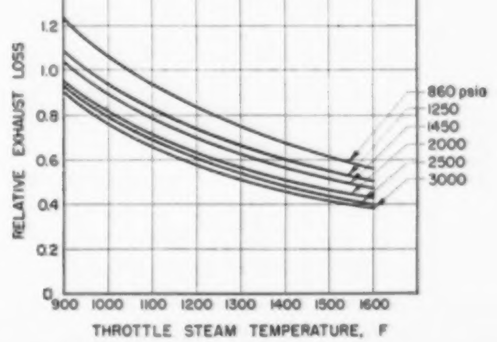


Fig. 8—Relative exhaust loss vs. throttle temperature

[3] "Relative Engine Efficiencies Realizable From Large Modern Steam-Turbine-Generator Units," by G. B. Warren and P. H. Knowlton, A.S.M.E. *Trans.*, vol. 63, February 1941.

EVALUATION OF HEAT RATE DIFFERENCES

ADDITIONAL JUSTIFIED INVESTMENT PER 100,000 KILOWATTS
OF CAPACITY OVER COST OF 1250 PSIA-950 F PLANT
COST FIGURES IN THOUSANDS OF DOLLARS

	Coal Cost Cents per Million Btu	1250 Psia Annual Plant Factor--Per Cent 50	60	70	80	1450 Psia Annual Plant Factor--Per Cent 50	60	70	80	2000 Psia Annual Plant Factor--Per Cent 50	60	70	80
	10	- 61	- 74	- 86	- 98	- 16	- 19	- 22	- 25	62	74	86	99
	15	- 92	- 110	- 129	- 147	- 24	- 28	- 33	- 38	93	111	130	148
900 F	20	- 123	- 147	172	- 196	- 32	- 38	- 44	- 50	123	148	173	198
	25	- 153	- 184	- 215	- 245	- 39	- 47	- 55	- 63	154	185	216	247
	30	- 184	- 221	- 258	- 295	- 47	- 57	- 66	- 76	185	222	259	296
	35	- 215	- 258	- 301	- 344	- 55	- 66	- 77	- 88	216	259	303	346
	10	59	71	83	95	108	129	151	172	183	220	256	293
	15	89	106	124	142	162	194	226	259	275	330	384	439
1000 F	20	118	142	166	189	215	259	302	345	366	439	513	586
	25	148	177	207	237	269	323	377	431	458	549	641	732
	30	177	213	248	284	323	388	453	517	549	659	769	879
	35	207	248	290	331	377	453	528	603	641	769	897	1,025
	10	169	203	237	270	213	255	298	341	289	347	405	463
	15	254	304	355	406	319	383	447	511	434	520	607	694
1100 F	20	338	406	473	541	426	511	596	681	578	694	809	925
	25	423	507	592	676	532	639	745	851	723	867	1,012	1,156
	30	507	609	710	811	639	766	894	1,022	867	1,041	1,214	1,388
	35	592	710	828	947	745	894	1,043	1,192	1,012	1,214	1,416	1,619
	10	259	311	362	414	306	367	428	489	381	458	534	610
	15	388	466	544	621	459	550	642	734	572	687	801	916
1200 F	20	518	621	725	828	611	734	856	978	763	916	1,068	1,221
	25	647	777	906	1,035	764	917	1,070	1,223	954	1,144	1,335	1,526
	30	777	932	1,087	1,243	917	1,101	1,284	1,467	1,144	1,373	1,602	1,831
	35	906	1,087	1,268	1,449	1,070	1,284	1,498	1,712	1,335	1,602	1,869	2,136
	10	427	513	598	684	470	564	659	753	541	649	757	865
	15	642	769	898	1,026	706	847	988	1,129	811	974	1,136	1,298
1400 F	20	855	1,026	1,197	1,368	941	1,129	1,317	1,505	1,082	1,298	1,515	1,731
	25	1,069	1,282	1,496	1,710	1,176	1,411	1,646	1,881	1,352	1,623	1,893	2,164
	30	1,282	1,539	1,795	2,052	1,411	1,693	1,976	2,258	1,623	1,947	2,272	2,596
	35	1,496	1,795	2,095	2,394	1,646	1,976	2,305	2,634	1,893	2,272	2,651	3,029
	10	564	676	789	902	605	726	947	968	675	810	946	1,081
	15	846	1,015	1,184	1,353	908	1,090	1,271	1,453	1,013	1,216	1,418	1,621
1600 F	20	1,127	1,353	1,578	1,804	1,211	1,453	1,695	1,937	1,351	1,621	1,891	2,161
	25	1,409	1,691	1,973	2,255	1,513	1,816	2,118	2,421	1,688	2,026	2,364	2,702
	30	1,691	2,029	2,367	2,706	1,816	2,179	2,542	2,905	2,026	2,431	2,837	3,242
	35	1,973	2,367	2,762	3,157	2,119	2,542	2,966	3,390	2,364	2,837	3,309	3,782

	Coal Cost Cents per Million Btu	2500 Psia Annual Plant Factor--Per Cent 50	60	70	80	3000 Psia Annual Plant Factor--Per Cent 50	60	70	80
	10	97	116	136	155	106	127	148	169
	15	145	174	203	232	158	190	222	253
900 F	20	194	232	271	310	211	253	296	338
	25	242	290	339	387	264	317	369	422
	30	290	348	407	465	317	380	443	507
	35	339	407	474	542	369	443	517	591
	10	227	272	318	363	242	291	339	388
	15	340	408	476	544	363	436	509	581
1000 F	20	454	544	635	726	484	581	678	775
	25	567	681	794	907	606	727	848	969
	30	681	817	953	1,089	727	872	1,017	1,163
	35	794	953	1,112	1,270	848	1,017	1,187	1,356
	10	342	410	478	546	357	428	499	570
	15	512	615	717	820	535	642	749	856
1100 F	20	683	820	957	1,093	713	856	998	1,141
	25	854	1,025	1,196	1,367	891	1,070	1,248	1,426
	30	1,025	1,230	1,435	1,640	1,070	1,284	1,497	1,711
	35	1,196	1,435	1,674	1,913	1,248	1,497	1,747	1,997
	10	430	516	602	688	459	551	643	735
	15	645	774	903	1,032	689	827	965	1,103
1200 F	20	860	1,032	1,204	1,376	919	1,103	1,287	1,470
	25	1,075	1,290	1,505	1,720	1,149	1,378	1,608	1,838
	30	1,290	1,548	1,807	2,065	1,378	1,654	1,930	2,205
	35	1,505	1,807	2,108	2,409	1,608	1,930	2,251	2,573
	10	593	712	830	949	626	752	877	1,002
	15	890	1,067	1,245	1,423	940	1,127	1,315	1,503
1400 F	20	1,186	1,423	1,661	1,898	1,253	1,503	1,754	2,004
	25	1,483	1,779	2,076	2,372	1,566	1,879	2,192	2,505
	30	1,779	2,135	2,491	2,847	1,879	2,255	2,631	3,006
	35	2,076	2,491	2,906	3,321	2,192	2,631	3,069	3,507
	10	724	869	1,014	1,158	754	905	1,055	1,206
	15	1,086	1,303	1,520	1,738	1,131	1,357	1,583	1,809
1600 F	20	1,448	1,738	2,027	2,317	1,508	1,809	2,111	2,412
	25	1,810	2,172	2,534	2,896	1,884	2,261	2,638	3,015
	30	2,172	2,607	3,041	3,475	2,261	2,714	3,166	3,618
	35	2,534	3,041	3,548	4,055	2,638	3,166	3,694	4,221

Allowances for miscellaneous auxiliary power demands were included in all cases.

Boiler Efficiency: A boiler manufacturer indicated that even with feedwater temperatures approaching 600 F on these high-pressure high-temperature boilers, the exit gas temperature could be held to about 300 F. Assuming pulverized-coal-fired boilers operating with an average grade of coal, it was estimated that boiler efficiencies would vary from 87.7 to 88.8 per cent, depending upon the feedwater temperature.

Turbine Heat Rates: Turbine heat rate differences represented in Fig. 4, which are based on the practical values, were used in the evaluation.

Annual Fixed Charges: An annual fixed charge rate of 10 per cent was assumed in computing the justified additional investments. The investment figures shown in the table may be adjusted for fixed charge rates.

High Steam Temperature Vs. Resuperheating

It is of interest to compare the heat rate gains due to higher steam temperatures with those made possible by resuperheating. An approximate comparison may be made directly by comparing gains shown in Fig. 4 of this paper with various charts by Harris and White, and by Reynolds.[4] These two references do not exactly agree on the gain due to reheat, presumably due to differences in the assumptions made by the authors in setting up their calculations.

The assumptions made by Harris and White differ from those made in this paper on two points:

1. The boiler-feed temperatures are lower in the present case than were assumed by Harris and White. This makes a minor change in the gain due to reheat.
2. The heat rate gains in the present Fig. 4 are on the basis of a fixed percentage exhaust loss, whereas Harris and White assumed, in calculating their reheat gain, that there non-reheat machines had a 4.5 per cent exhaust loss and their reheat machine a lesser exhaust loss obtainable from keeping the same exhaust-end size and the same rating for their reheat turbine. The assumptions made by Reynolds regarding the exhaust loss as between reheat and non-reheat are not stated.

It is possible in the present case to show the effect of considering either of two cases:

(*a*) The use of a *constant percentage exhaust loss,* regardless of throttle steam conditions. In this case, as we proceed to higher throttle temperatures, the exhaust size per unit rating decreases, and the gains expected are as shown in Fig. 4.

(*b*) The use of a *constant size of exhaust per unit rating,* regardless of throttle steam conditions. In this case, the percentage exhaust loss decreases when throttle temperature increases, and a further gain results from this, over and above the gains indicated in Fig. 4. The magnitude of this change in exhaust loss can be calculated by the use of Fig. 8, if the exhaust loss at a base condition is known. For any two sets of steam conditions shown on Fig. 8, the ratio of exhaust loss percentage is the ratio of the ordinates therein. An example will illustrate:

Assume a base condition of 1450 psia, 1000 F, and compare heat rate gains due to: (1) Reheat to 1000 F at 450 psia reheat pressure; (2) raising throttle temperature to 1250 F, non-reheat.

[4] "Reheating in Steam Turbines," by R. L. Reynolds, A.S.M.E. *Trans.,* vol. 71, No. 6, August 1949.

The answer to (1) can be read from Harris and White (Fig. 11) as 5.2 per cent gain due to reheat, on the basis of constant exhaust size per unit rating.

The answer to (2) is given here in two parts, corresponding to the foregoing Items (*a*) and (*b*):

(*a*) For constant percentage exhaust loss from Fig. 4, the gain at 1450 psia, 1000 F is 2.4 per cent and at 1450 psia, 1250 F it is 7.9%. This gives a net gain of

$$\frac{7.9 - 2.4}{1 - 0.024} = 5.6 \text{ per cent}$$

(*b*) For constant size of exhaust per unit rating, the relative exhaust loss factors from Fig. 8 are as follows:

1450 psia, 1000 F	0.89
1450 psia, 1250 F	0.66

Revised exhaust loss percentage from the assumed 4.5 per cent is

$$4.5 \times 0.66/0.89 = 3.4 \text{ per cent}$$

Additional heat rate gain due to exhaust loss correction is

$$4.5 - 3.4 = 1.1 \text{ per cent}$$

Total gain due to temperature increase is

$$5.6 + 1.1 = 6.7 \text{ per cent}$$

It will be noted that the additional gain due to a change in the exhaust loss with fixed exhaust size depends directly upon the magnitude of the exhaust loss itself.

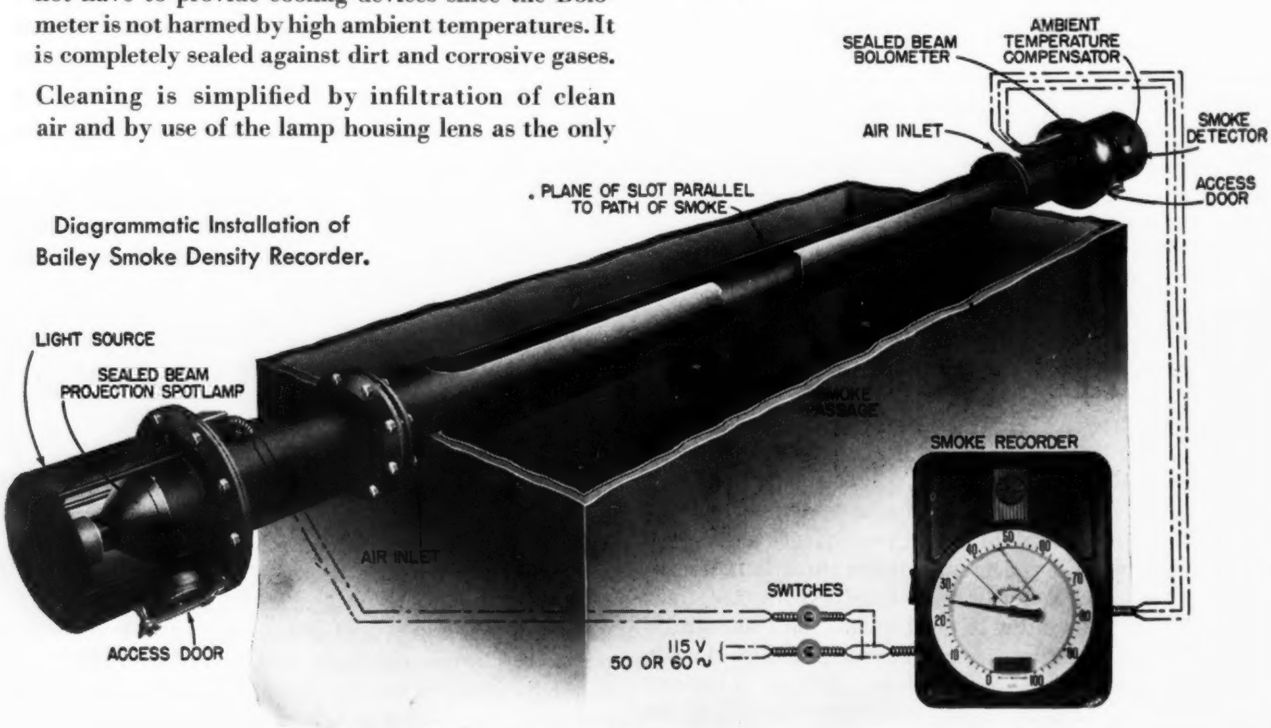
In comparing heat rate gains due to higher steam temperatures with those made possible by resuperheating as presented by Harris and White, an exact evaluation should include the exhaust loss correction described in Item (*b*).

Conclusion

The curves for practical performance in Fig. 4 exhibit trends similar to those for the theoretical performance in Fig. 3. However, it will be seen that the practical cycle shows less improvement with increasing pressure, and more improvement with increasing temperature, than the theoretical cycle. This relationship results from the fact that the practical turbine loses in efficiency with increasing pressure, and gains in efficiency with increasing temperature, while the theoretical turbine is assumed to be always 100 per cent efficient.

Use of steam temperatures in power plants above the present maximum of 1050 F will depend on the development of suitable materials for the higher temperatures, and on the design of acceptable equipment, particularly boilers and turbines, at a cost commensurate with the thermal gains to be realized. Since the costs of such materials and equipment are not presently available, this paper has presented only the total investment which could be justified on the basis of the thermal gains expected at the higher steam conditions.

The authors wish to express their appreciation for the advice and assistance of Mrs. Jean Harrison Higley and Mr. E. E. Harris of the General Electric Company, Mr. A. E. Raynor of the Babcock & Wilcox Company and Messrs. H. A. Wagner and H. S. Walker of The Detroit Edison Company. The data on steam in the foregoing calculations are from "Thermodynamic Properties of Steam," by Keenan and Keys.



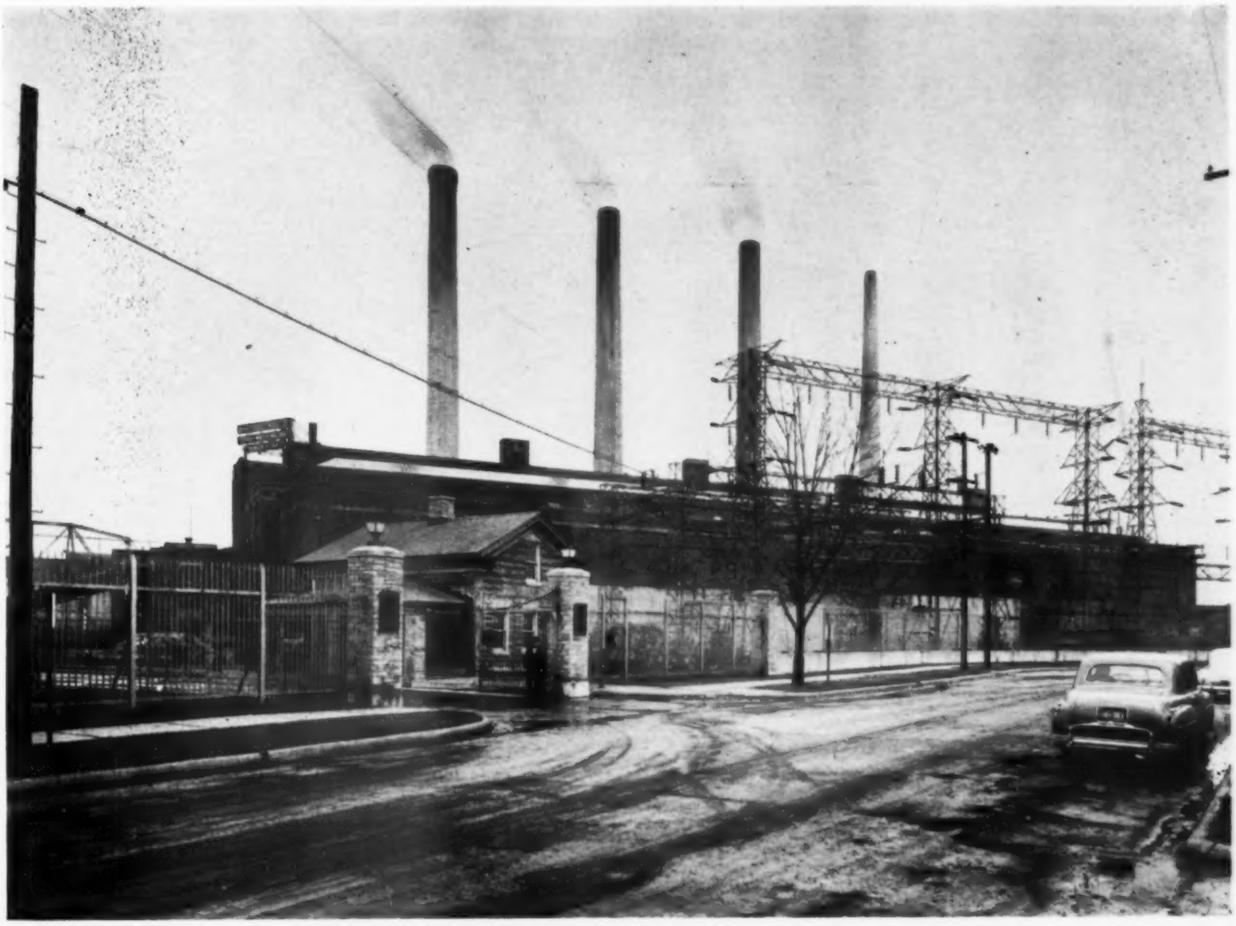
Diagrammatic Installation of Bailey Smoke Density Recorder.

1949 PORT WASHINGTON EXPERIENCES

Starting of Unit No. 4 on August 25; correction of Unit No. 2's high-pressure heater tube leaks; development of multiple steam-heating of turbine cylinder bolts; burning of 50 per cent western Kentucky coal of 2040 F ash-fusion temperature; simultaneous outage of two units on two occasions after Unit No. 4 operated (although the total forced outage time of all units for the year was only 0.5 per cent); dismantling of turbines Nos. 1 and 2 after 32,000 average operating hours each; zero radiant superheater and reheater tubing replacements; and no screen tube corrosion experiences: these appear the high points of 1949 experiences. During its first four months of operation Unit No. 4 averaged 9939 Btu per net kw hr.

BECAUSE turbine No. 4 at Port Washington Station of the Wisconsin Electric Power Company was ready for operation about two weeks before the boiler, it was tuned-up by using steam from boiler No. 3 through two long auxiliary lines of only $1^1/_2$ in. inside diameter. Starting steam flow was minimized by establishing full 29 in. vacuum before accelerating above turning-gear speed, as permitted by employment of steam seals at all glands. The booster air-removal steam jet could not have been supplied while the turbine was at speed, which further urged early attainment of full vacuum. Despite this seemingly small steam capacity to operate the turbine, the unit was run 10 per cent overspeed, with a reserve margin. Modern steam units require surprisingly small steam flow to sustain rated speed.

Thick steel tubes in the six high-pressure heaters of Unit No. 2 had caused much trouble due to leaks at the tube-sheet joints, because rolling and welding had not been tight, and leaks also occurred at high condensate velocity points because of "smooth" corrosion. Re-



Exterior view of Port Washington Station of four units

Table 1

OUTPUT AND ECONOMY DATA

Unit No.	Period (Incl.)	Net Output 10^6 Kw-hr.	B.t.u./Kw-hr. Net
1	1949	463.072	11,036
	1935*-49	6,248.275	10,852
2	1949	486.479	10,722
	1943*-49	3,156.472	10,603
3	1949	562.288	10,087
	1948*-49	679.839	10,089
4	1949*	184.113	9,939
Plant	1949	1,695.951	10,510

*Nov. 22, Oct. 27, Oct. 5, and Aug. 25 startings, respectively.

placement with copper-nickel tubes while operating, by displacing each heater in turn with a feedwater bypass pipe, has regained high heat transfer and improved economy almost 1 per cent, without experiencing any leak.

A day's outage time was saved by the simultaneous heating with saturated steam of groups of four high-pressure cylinder bolts while four adjacent ones were being prepared for heating. Steam pressure was raised in the bolts during a 15-minute period at a calculated controlled rate that heated them uniformly and exactly enough to obtain the desired stress upon cooling, without any "cut-and-try." Extension measurements and the amount of turning of the nuts confirmed this proper stressing. The process had been devised to avoid unfavorable stressing of the bolts, but its greatest merit proved to be its resulting rapidity and ease. It will be further developed to heat twelve bolts simultaneously, six in a group on each side of the cylinder, for both dismantling and tightening.

Western Kentucky coal with relatively low ash-fusion temperature (2040 F) was often preferred to Eastern coal of equally high sulfur and almost double the ash quantity. The relatively large furnace-cooling surface, resulting in average furnace temperatures that do not exceed ash-softening temperatures, is maintained relatively clean, consistently, by careful operation during normal runs of several months continuously under load. The first three units averaged slightly less than four stops during the year.

Not until Unit No. 4 was in service did Port Washington experience simultaneous outage of two units. Failure of one of its generator leads early one Saturday morning took Unit No. 4 out on differential relay while Unit No. 3 was out for its fall inspection. On another occasion, testing of Unit 4's fault-bus relays took out Unit No. 1 on a Thursday afternoon for 19 minutes, while Unit No. 4 was similarly out on schedule. Fortunately no service impairment resulted. The need of high-pressure steam cross-connections has not as yet occurred.

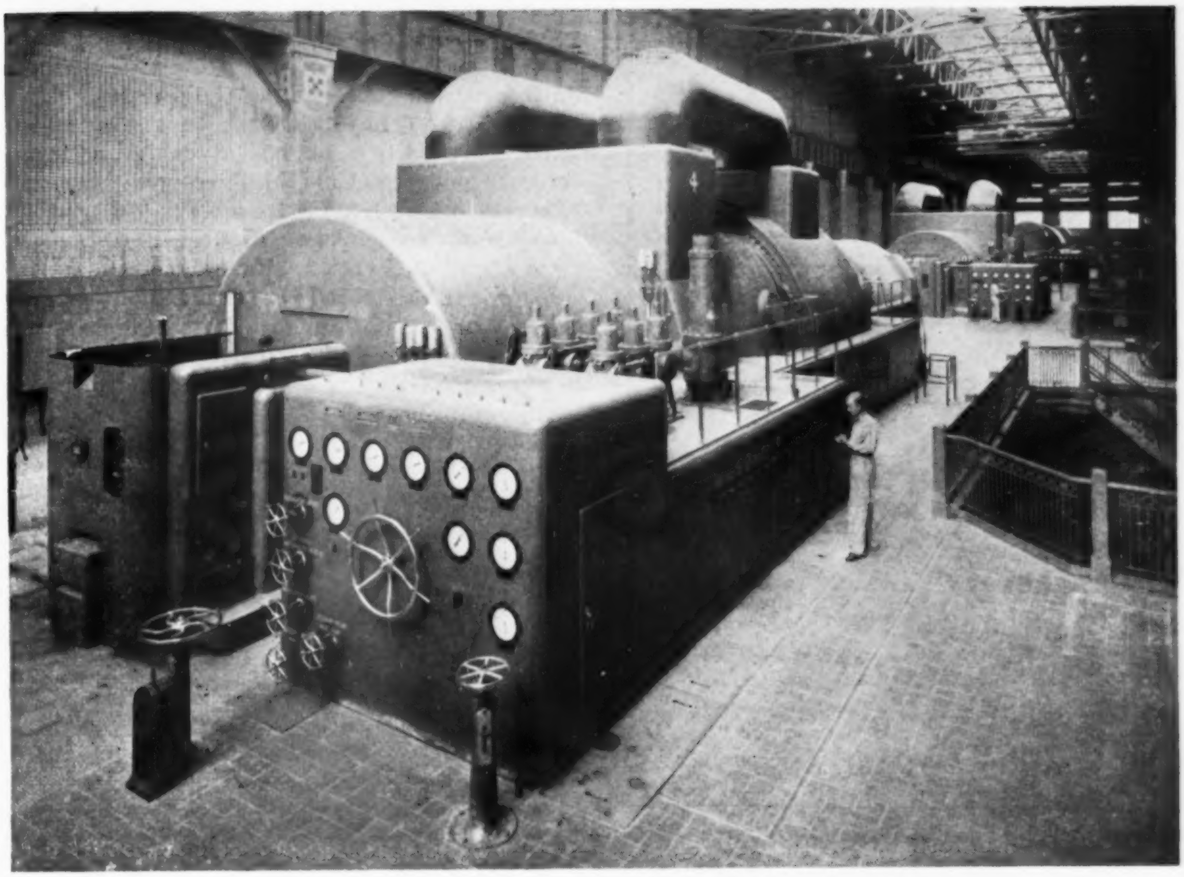
A forced outage time of 0.8 per cent has been averaged by the Port Washington units over the last three years; 0.3 per cent for the boilers and 0.5 per cent for the turbines. Inspection of the tabulations describing outages published in this and previous January issues of COMBUSTION will tell the nature of the forced outages. Early corrosion trouble on boiler No. 1 and No. 1 turbine blading

Table 2

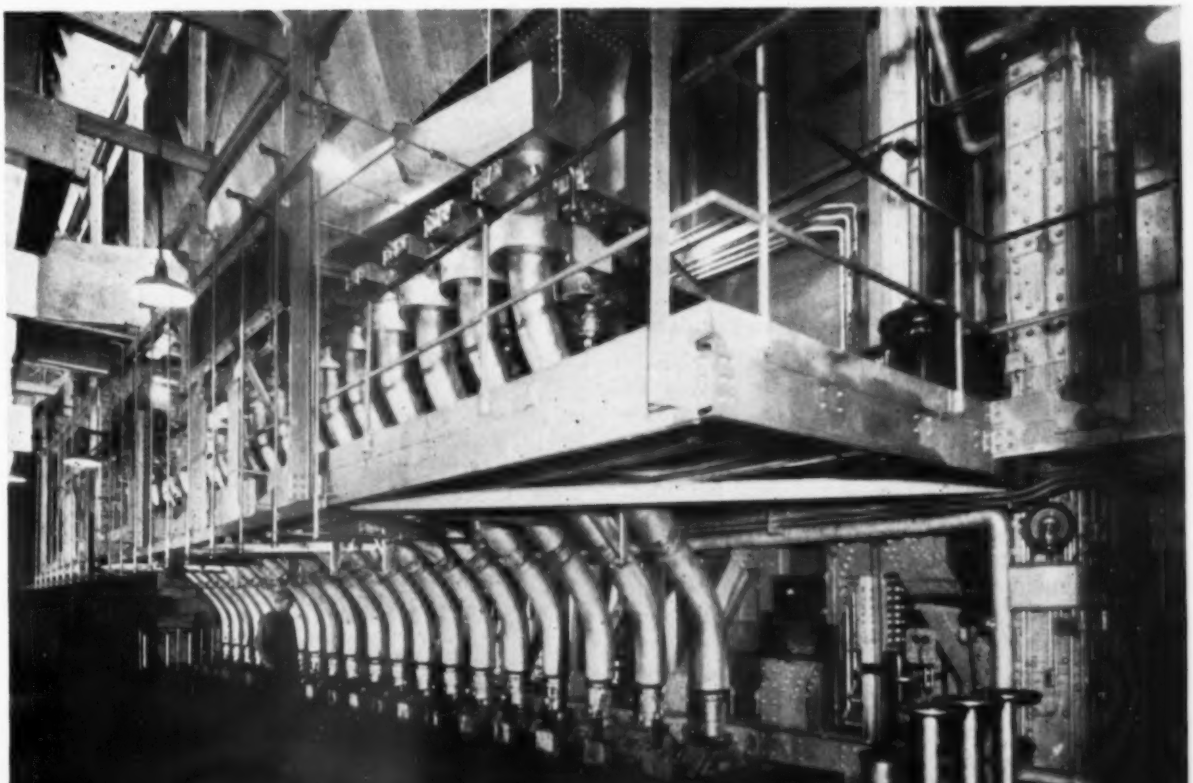
USE, AVAILABILITY AND FORCED OUTAGE FACTORS

Unit	Period		USE Serv. Hr. / Period Hr.	HOURLY-OUTPUT Av.Hrly.Output / Rated " "	ANNUAL-OUTPUT Annual Output / " Rated "	DEMAND Demand Hr. / Period "	DEMAND-AVAIL. Service Hr. / Demand Hr.	AVAILABILITY 100-Repair Hr. / Period Hr.	FORCED OUTAGE TIME Forced Outage Hr.* / Period Hr.
1	1949	Blr.	94.9	68.3	64.9	99.4	95.5	95.5	0
		Turb.	94.9	73.8	70.0	99.9	95.0	95.0	0
		Unit	94.9	73.8	70.0	100	95.0	94.9	0
	1936-49	Blr.	90.1	67.5	60.9	93.6	96.4	94.6	0.7
	(14 Yr.)	Turb.	90.1	74.0	66.8	97.3	92.6	93.1	2.0
		Unit	90.1	74.0	66.8	99.3	90.7	90.7	2.7
2	1949	Blr.	94.8	71.3	67.6	99.5	95.4	95.4	0.2
		Turb.	94.8	77.3	73.3	99.8	95.1	95.1	0
		Unit	94.8	77.3	73.3	100	94.8	95.0	0.2
	1943**- 1949	Blr.	93.0	76.3	70.4	95.9	97.1	95.8	0.1
	(6.2 Yr.)	Turb.	93.3	82.4	77.0	98.1	95.1	94.0	0.1
		Unit	93.3	82.4	77.0	100	98.1	93.3	0.2
3	1949	Blr.	94.9	78.1	74.1	99.1	95.7	96.0	0.3
		Turb.	94.9	89.0	84.5	99.6	95.3	95.6	0.8
		Unit	94.9	89.0	84.5	100	94.9	95.2	1.2
	1948**- 49	Blr.	94.0	77.7	73.0	97.3	96.6	96.8	0.3
	(1.2 Yr.)	Turb.	94.0	88.6	83.3	99.7	94.3	94.6	1.3
		Unit	94.0	88.6	83.3	100	94.0	94.2	1.6
4	1949**	Blr.	95.1	72.5	68.9	98.8	96.2	96.3	0
	(0.3 Yr.)	Turb.	95.1	86.2	81.9	98.8	96.2	96.3	0
		Unit.	95.1	86.2	81.9	100	95.1	95.1	1.2
Plant	1949	Blrs.	94.9	72.5	68.9	99.3	95.6	95.7	0.2
		Turbs.	94.9	80.7	76.5	99.7	95.2	95.3	0.2
		Units	94.9	80.7	76.5	100	94.9	95.0	0.5
	1936-49	Blrs.	91.2	70.6	64.4	94.5	96.6	95.1	0.5
		Turbs.	91.3	77.4	70.8	97.7	93.5	93.5	1.4
		Units	91.3	77.4	70.8	99.5	93.1	91.7	1.9

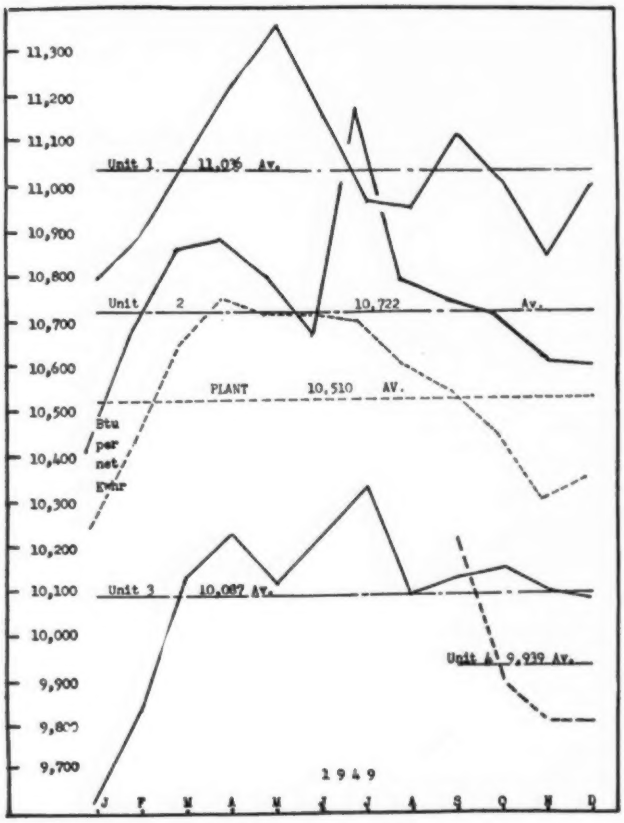
*Forced outage time due to plant equipment failure, including immediate and delayed forced outages.
**October 27, October 5, and August 25 startings, resp. Oper. Hr. used to weight plant data.



The turbine room contains four 80,000-kw turbine-generators



Boiler front showing pulverized-coal piping to vertical burners



Comparative heat rates of four units

and generator trouble impair the long-time record of Table 2. A forced outage time average of 0.2 per cent for Unit 2 during 6.2 years is an excellent goal for all units. Forced outage time is perhaps a more vital statistic than availability, which for the same three years averaged 96.2 per cent for the boilers, 96.3 per cent for the turbines and 95.3 per cent for the plant. Operating time of each unit during the last three years was close to 95 per cent.

Turbines Nos. 1 and 2 had operated approximately four years each before being dismantled on schedule during the low-load holiday period in July. A moderate amount of silica scale was found near the atmospheric pressure point (250 F), notwithstanding the fact that boiler-water silica had been maintained throughout most of the period at the favorably low value of about 0.9 ppm. Efficiency loss for the last couple of months before dismantling indicated well over 1 per cent loss due to these and the moderate water-soluble deposits that occur on long runs, despite about 55 ppm. boiler-water solids.

Not since 6 per cent of the radiant reheater tubing on boiler No. 1 was replaced in 1946 by in-place welding of new straight lengths during a scheduled outage, has there been any replacement of superheater or reheater tubing. There have been a few welds made during scheduled outages, due to minor shop-weld leaks or to small "firecracks" on the face of radiant reheater tubes, but otherwise the superheating equipment has been practically without maintenance. Superheating plus reheating duty is 65 per cent of evaporating duty in Units Nos. 1 and 2, and 70 per cent in Units Nos. 3 and 4. Of the total boiler-unit heat input, they account for 32 per cent and 35 per cent, respectively.

Unit No. 3 has averaged 10,087 Btu per net kilowatt-hour during the year, or 0.9 per cent more than the 10,000 figure predicted last year. Unit No. 4 appears to be able to stay within the four-figure range, barring unexpected trouble, for it averaged 9939 Btu per net kilowatt-hour during its first four months of operation. Longer air heater plates, omission of the upstream intercepting valve, and slightly higher boiler pressure are the principal causes of gains over Unit No. 3's economy. The increased heat rate of Unit No. 1 reflects lower loads, for this turbine has only two inlet valves. Its incremental heat rates exceed those of the later units. The plant's average heat rate for 1949 was 10,510 Btu per net kilowatt-hour.

The four Port Washington units have generated two-thirds of the system output during recent months, and on Sundays are operating essentially alone with minimum simultaneous loads of 20,000 kw each. That their loading is not at the expense of system economy is indicated by system heat consumption during October, November and December 1949, of 11,893 Btu per net kilowatt-hour.

Unit No. 5 is scheduled to operate at the end of 1950, or early in 1951.

Table 3

OPERATING PERIODS AND REASONS FOR OUTAGES, 1949

Unit	Operating Period: No.	Start	Finished	Hrs. Run	KWH Generated	Outage: Hrs.	Reason
1	81	12-27-48*	6-24-49	4317.00	256,418,000	358.65	General inspection and repair boiler and turbine. Install new type blading, LP spindle, HP spindle, and cylinder (scheduled).
	82	7-9-49	9-15-49	1627.10	97,439,000	.32	Error in testing fault bus (forced).
	83	9-15-49	10-19-49	823.10	49,103,000	83.70	General inspection of boiler, turbine and auxiliaries (scheduled).
	84	10-23-49	12-26-49**	1550.13	87,883,000		
	Total	12-27-48	12-26-49	8317.33	490,843,000	442.67	
	Total	11-22-35	12-26-49	113383.43	6,599,075,000	12,192.57	
*This period started 10-24-48. **Still in operation.							
2	31	12-27-48*	4-29-49	2974.00	185,273,000	21.50	Leaky plug radiant superheater header No. 2 side (delayed forced).
	32	4-30-49	7-10-49	1695.92	105,736,000	345.96	General inspection and repair boiler and turbine. Replace blading, H.P. & L.P. cylinder and spindle (scheduled).
	33	7-24-49	10-26-49	2256.99	139,660,000	83.86	General inspection and repair turbine and boiler (scheduled).
	34	10-30-49	12-26-49**	1381.77	83,240,000		
	Total	12-27-48	12-26-49	8308.68	513,909,000	451.32	
	Total	10-27-43	12-26-49	50787.32	3,332,135,000	3585.84	
*This period started 11-7-48. **Still in operation.							
3	3	12-27-48*	3-27-49	2167.00	155,888,000	28.02	Soot blower cutting of boiler tubes (delayed forced).
	4	3-28-49	6-10-49	1788.48	128,794,000	189.85	General inspection and repairs boiler and turbine (scheduled).
	5	6-18-49	6-23-49	119.43	8,210,000	7.05	Leaky bonnet, feed water tie valve (delayed forced).
	6	6-24-49	9-30-49	2370.45	170,484,000	152.92	General inspection and repairs (scheduled).
	7	10-7-49	10-13-49	159.25	10,392,000	67.58	Leak front end of turbine joint (delayed forced).
	8	10-16-49	12-26-49**	1709.97	118,138,000		
	Total	12-27-48	12-26-49	8314.58	591,906,000	445.42	
	Total	10-5-48	12-26-49	10097.11	715,742,000	634.84	
*This period started 10-31-48. **Still in operation.							
4	1	8-25-49*	9-14-49	478.03	24,091,000	110.48	Initial inspection turbine and boiler (scheduled).
	2	9-19-49	10-15-49	624.97	45,658,000	35.25	Cable on "B" phase generator lead failed (forced).
	3	10-16-49	12-26-49**	1713.30	124,362,000		
	Total	8-25-49	12-26-49	2816.30	194,111,000	145.73	
*Initial operation of this unit. **Still in operation.							

Mechanical Vacuum Pumps in Central Station Operation

By R. C. WEBSTER
Kinney Manufacturing Co.

A brief history of the applications of this type of pump is followed by a discussion of its adaptability to power-house service. The pump is described, its characteristics given, and operating performance in a new large high-pressure central station is reviewed.

THE mechanical rotary vacuum pump has been built in industrial sizes for about thirty years. At first it was used principally in the manufacture of incandescent lamps, but for the last twenty years it has had a place in industrial processing work as well. Its ability to handle air-water vapor was recognized in the late 20's when it was applied to the drying of the electric power cables. Later it was used for desiccation work in pharmaceuticals; then came the drying of foods, penicillin, refrigerator units, priming centrifugal pumps, blood plasma and many other applications of vacuum drying. All of these applications demonstrated the ability of the pump to handle water vapor from a pressure ranging from atmospheric to within a few thousandths of a millimeter of mercury, absolute pressure. At present there are several thousand of these pumps of all sizes operating on processes where water vapor and air are present in varying amounts, some of which have been in continuous service for as long as twenty years.

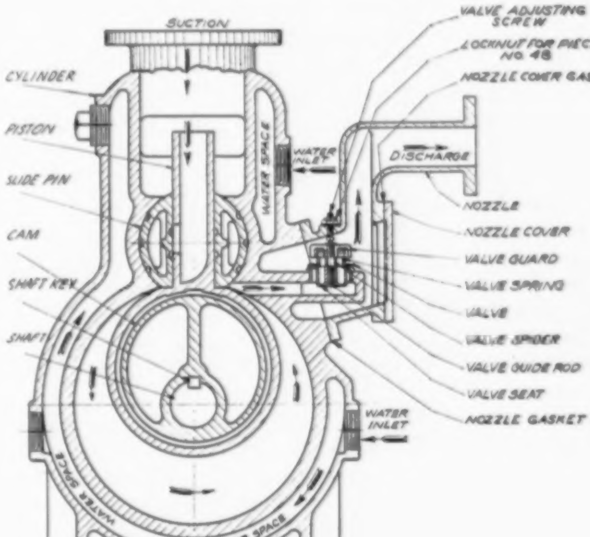
In the last few years there has been an increasing interest on the part of power-house operators and designers in the use of mechanical rotary vacuum pumps on steam condensers in place of steam jet air ejectors. This interest has increased to the extent that there are now a number of units in operation, several more under construction and more under consideration.

With the ever-increasing use of high-temperature high-pressure steam, there seems to be a desire to eliminate steam lines to accessory equipment. Steam jets normally operate at reduced pressures, and high maintenance in the pressure-reducing equipment is sometimes encountered. There is also a desire on the part of some operators for automatic equipment which can be operated from a single control panel. In this respect, mechanical rotary vacuum pumps can be started from the control room by merely pressing a button. There are other operators who experience ammonia contamination which can be entirely eliminated with the rotary vacuum pump. For those who are interested in automatic control, the rotary vacuum pump is capable of air removal over the entire range of pressures, from atmospheric to the lowest pressure desired. Some are interested in the ability of the rotary vacuum pump to pick up a load quickly and the characteristics of the pump lend it admirably to this condition. Others use the pump to test a new unit for air leakage before steam is available.

To date the mechanical rotary vacuum pump has not been offered for use with steam generating units operating at less than 1000 psi, inasmuch as the initial costs involved make it economically undesirable to consider this type for lesser pressures. However, the cost of the mechanical vacuum pump remains the same regardless of steam pressure, whereas the steam-jet air pump with its auxiliary equipment will vary with the steam pressure.

Description of Pump

The pump normally used in this type of work, as shown in the cross-section, is liquid-sealed and of the duplex type, consisting of a water-jacketed cylinder in which there are two eccentrically mounted rotors or pistons, set 180 deg apart on a horizontally rotating shaft. The shaft is mounted on two bronze bearings included in the cylinder heads, and the shaft extends through one of the cylinder heads for the drive which is either through a V-belt from a high-speed motor or by direct connection to a slow-speed motor.



Section through pump

This type of pump requires some means of sealing and lubrication of the various working parts. For this pur-

pose a vertical cylindrical tank is supplied and is attached to the pump discharge. This tank, which is generally called a separator tank, is about one-third full of the sealing and lubricating medium, which is supplied at the rate of a few gallons per minute. It enters the pump through the bearings and under about 15-lb pressure. Inasmuch as this sealing and lubricating medium is under pressure, it enters the pump cylinder at a constant rate where it lubricates and seals the working parts and passes out through the discharge of the pump and back into the separator tank. In the separator tank there is a series of baffles whose function it is to separate the gases from the liquid. When large volumes of gases are passing through the separator tank, small amounts of the sealing medium are carried along with the discharged gases. In order to save this sealing medium an accessory separator is placed at the far end of the discharge line. This auxiliary separator is called a "whirl type or cyclonic" separator and functions much the same as a mechanical dust collector.

At the discharge side of the pump there is a series of poppet valves through which the discharged gases pass. These act as check valves in preventing the back flow of the gases and sealing medium through the pumping cycle.

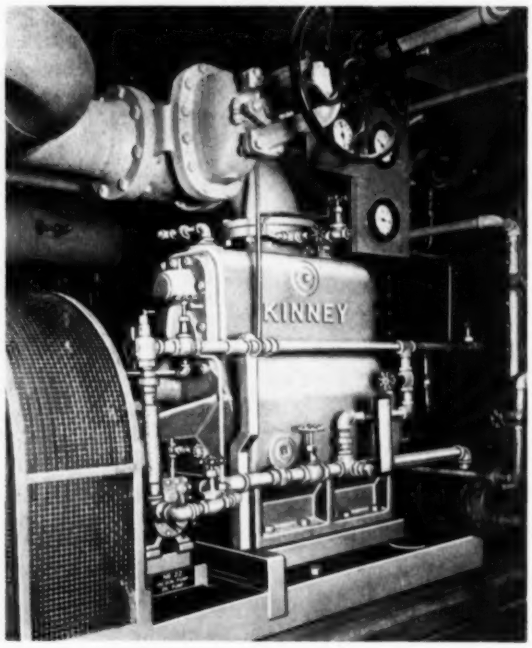
Inasmuch as this type of vacuum pump is liquid-sealed it is necessary to select a liquid of sufficiently low vapor pressure so that it will not vaporize under the reduced pressures encountered, yet high enough in viscosity to seal the pump. This liquid must also have lubricating value to lubricate properly any internal bearings. Normally oil is used for this purpose and for the type of vacuum pump under discussion, the oil should have a viscosity of about 400 SSU at 122 F, and should be non-emulsifying.

Following months of tests, an installation was made on a new 100,000-kw unit. This twenty-four stage turbine operates at a speed of 3600 rpm, utilizing steam at 1275 psi and 970 F. The 55,000-sq ft condenser is cooled by water drawn from an adjacent river. The vacuum-pump unit consists of two pumps each independently driven and the air piping arranged so that either or both could be used. A single discharge line was connected to both pumps through a manifold and one cyclonic, or whirl-type, separator used at the far end of the line. A flowmeter was placed in the discharge line so that the amount of air passing through the pumps could be recorded.

Operation

The initial starting was a little slower than usual, and with both vacuum pumps in operation it was necessary to open a 3-in. line to atmosphere to prevent the vacuum from running too high. As soon as the turbine had reached sufficient speed to close the seals, the 3-in. valve was closed and the pressure reached 1 in. Hg abs very promptly.

Adjustments to the turbine made it necessary to stop the unit several times before being placed on the line. During the several starts, pump-down tests were made with one and two pumps in operation. With one pump in operation a vacuum of 17 in. Hg could be reached in twenty minutes and with two pumps in operation ten minutes were required to reach the same pressure. Normally one pump is in operation with the other acting as a standby and both pumps are operated from the control room. When the control engineer notices the flowmeter indicating over 4 cfm, he starts the standby unit and runs it until the air leakage has been corrected. In this particular installation the air leakage was around 2 cfm.



Installation in large post-war electric generating station

After thirteen months of continuous operation the unit was shut down and during the shutdown the pumps were examined. No visible wear could be detected in either of the units. A thin protective film of the oil covered the inside of the pumps and prevented the formation of any rust which would be normally expected with the relatively large amount of water passing through the pumps.

With only one pump in operation, the idle pump is holding the vacuum. This is accomplished by means of the discharge valves in the pump which are nothing more than liquid-sealed check valves. It is also necessary to shut off the liquid-seal lines when the pump is inoperative, otherwise the seal would continue to flow from the separator tank. This is accomplished by means of an electric solenoid valve which opens and closes simultaneously with the starting and stopping of the pump motor.

The solenoid valves used on this equipment are of the packless type; but some operators do not favor this type of valve because there is no visible means of determining whether it is open or closed. To compensate for the omission of the electric solenoid valve and still maintain automatic operation, a small motor-driven pump is supplied which works simultaneously with the vacuum pump motor. This small pump operates against a relief valve set at about 45 psi and discharging into the seal lines. When the pump is stopped the closing of the relief valve functions the same as the electric solenoid valve.

As mentioned earlier, oil emulsifies when coming in contact with water vapor and would soon become useless when encountering the quantities of water prevalent in condenser applications. Heat will break down an oil emulsion, and therefore the pumps are operated at a temperature above the boiling point of water; that is, the water is taken into the pump as a vapor and goes out the

same way. The vacuum pump unit has a simple thermo-syphon cooling system on the water jacket. A thermostatically controlled electric heating element in the separator tank assists in maintaining an oil temperature of 240 F in the tank. Some operators prefer to circulate water from the hotwell through the vacuum pump water jacket, instead of using the thermo-syphon system. This is quite satisfactory and tends to keep the idle pump warm for easy starting. Pumps using heavy oil as the sealing medium would require more power to start, unless kept sufficiently warm to reduce the viscosity.

The oil used for this work has a viscosity of about 1600 SSU at 100 F and is as non-emulsifying as oil can be made. At an operating temperature of 240 F it has a viscosity of around 200 SSU. Tests made by the oil companies of this oil after many hours of operation indicated that it still maintained its color, viscosity and lubrication value.

Pump Characteristics

The characteristics of a mechanical rotary vacuum pump are such that it has a relatively constant pumping speed over the entire range of pressures encountered in condenser applications. The size under consideration for power-house work has a pumping speed of 550 cfm at 1 in. Hg abs. When operating on a surface condenser, the rotary vacuum pump handles an air-water-vapor mixture. This mixture has been cooled several degrees by passing through the air-cooler section of the condenser. The Heat Exchange Institute specifies this temperature depression as 7.5 deg F but a well constructed condenser will do much better; in fact some of the newer units will produce as much as 15 deg F or more.

Dalton's law of partial pressures states that each component of a mixture occupies the same volume as the entire mixture, but each component is at its own partial pressure and the sum of the partial pressures makes up the total pressure. Air and water vapor under the temperature and air conditions in a condenser obey Dalton's law with sufficient accuracy. Therefore, for a given volume of air at a constant temperature, $V_2 = V_1 \dfrac{P_1}{P_2}$.

Applying this to a condenser problem assume that the condenser pressure is 1 in. Hg abs, the saturation temperature 79 F, and the temperature depression is 7.5 deg F. Then by subtracting 7.5 deg from 79 F we have a temperature of 71.5 F at the outlet of the air cooler. The partial pressure of water vapor at 71.5 F is 0.78 in. Hg abs, which when substracted from the total pressure of 1 in. Hg abs gives the partial pressure of the air which in this case is 0.22 in. Hg abs. Applying this to Dalton's law we have $V_2 = 550 \dfrac{0.22}{30} = 4.0$ which means that this rotary vacuum pump can handle 4 cfm of atmospheric air and maintain an absolute pressure of 1 in. Hg absolute.

As stated, some of the newer condensers will produce a temperature depression as high as 15 deg F with cooling water temperatures in the region of 40 F. Under such conditions the pump would be capable of handling at atmospheric air leakage as high as 7.3 cfm. By operating two pumps slightly lower pressures may be obtained; but since the turbine efficiency curve versus back pressure in the condenser flattens out at about 1 in. Hg abs, there is little or no appreciable gain in this reduced pressure. However, the use of both pumps has its advantages in pumping down the system from atmospheric pressure. A recent test on a 55,000-sq ft condenser indicated a pressure of 10 in. Hg abs was obtained in 25 min with two mechanical rotary vacuum pumps in operation as against 30 min for the hogging jet.

In the final analysis it is the condenser that creates the vacuum and the vacuum pump merely removes the air which infiltrates the system. As manufacturing processes improve, condensers will be built with less air leakage and greater efficiency. The mechanical rotary vacuum pump by its versatility is capable of meeting not only the present but future conditions as well with a high degree of efficiency and economy.

Facts and Figures

Most high-pressure boilers are being operated with some form of phosphate control.

•

The longest steam-turbine blading now believed to be in commercial service is 23 in.

•

The world's installed hydroelectric capacity has been authoritatively estimated as about 65 million kilowatts.

•

Coal miners in the United States are calculated to have lost an average of $1200 in wages per man through stoppages and the short work week during 1949.

•

The United States possesses approximately one-half the world's coal reserves, according to surveys that have been made in different countries to date.

•

A survey recently completed by the Edison Electric Institute shows the reserve generating capacity of electric utilities, on a nationwide average, to be around 11½ per cent over the estimated December peak.

•

Glass fibers are in production with a tensile strength of 300,000 lb per sq in. and experimental glass fibers have, on test, been found to approach a million pounds per square inch.

•

Sodium sulfate and vanadium pentoxide are the principal constituents of ash from fuel oil that cause troublesome deposits on heat-absorbing surface, such as superheaters.

•

During the latter part of December the Bureau of Mines' coal hydrogenation demonstration plant at Louisiana, Mo., completed a continuous 7-week break-in run, converting coal tar oil and Wyoming coal to gasoline and other liquid fuels.

•

The Navy's two power trains, built during the war and each of 10,000-kw mobile capacity, have assisted a number of localities where temporary power shortages prevailed. The latest assignment for one of these trains is to supplement the power supply of the local utility in Mexico City.

•

They call it a "skyhook." It's a bolt about three feet long that may be used to support coal mine roofs from above, replacing the usual bulky timbering from below. The bolt, whose top end has a slit with a steel wedge placed in it, is driven up through a hole bored in the roof strata. Reaching the end of the hole, the wedge spreads the split bolt to anchor it firmly. When large washers are fastened tightly to the protruding lower end of the bolt, the skyhook holds the thin rock layers together to strengthen them after the manner of plywood, thus reducing the danger of cave-ins.

Invention and Sifting Out Engineering Facts

Following are excerpts from the Clayton Lecture delivered by E. G. Bailey* before the Institution of Mechanical Engineers in London, April 22, 1949, reprints of which have only lately become available. The relation between research and invention is explained, with a plea that research should be so conducted that more valuable inventions will be forthcoming. Incentives to invention are listed and steps in reducing an invention to practice are discussed. The author outlined his personal experience in developing combustion and steam generating equipment; and that leading up to the Bailey steam flow-air flow meter is here briefed as illustrative of the steps involved in bringing about a successful invention.

THE great increase in inventions began with the start of the Industrial Revolution in the Seventeenth Century when weaving, mining, metallurgy and other arts received great impetus. However, the real dynamic step was made when James Watt improved the efficiency of the steam engine and adapted it to the use of industry. An important factor in this rapid industrial development was the granting of patents for inventions, giving the inventor certain rights for a specified time. The foundations of modern patent law were established in England in 1624 by the Statute of Monopolies.

One invention led to another. The availability of the steam engine vitalized industry, and inventions were made not only in the generation and transmission of power, but in the equipment and uses made possible by its application. Inventions created industries; industries required engineering and management; and engineering improved efficiency, thereby reducing the costs and labor required to supply man's needs and wants.

We should now reverse the process and ask our engineers to concentrate on making industry as efficient as possible with the tools now available. Industry must plan and finance research to develop a more economic system, and research must lead to more worth-while inventions.

The incentive to improve is of greater importance, and yet is seemingly less prevalent than the ability to invent when some specific objective is to be attained. Incentives for invention may be generally classified as follows:

(*a*) Need for new products, processes, or services.

(*b*) Need to produce a product, perform an operation or process, or render a service with less labor, through the use of less or more readily available material in less time, or at less cost.

(*c*) The challenge to do something new for the gratification of accomplishment.

(*d*) The desire to benefit mankind.

The majority of inventions have been motivated by the first two incentives. Financial gain has been omitted as a primary incentive, because money alone is rarely the motivation for a useful invention, although the first two incentives listed usually lead to some financial gain, if successful.

In the United States today most engineering employees freely sign patent agreements whereby any inventions made by them, along the line of the employer's activities, are assigned to him. Some think that this tends to stifle the incentive to invent, but observations over a long period have shown that there is no withholding of ideas for this reason. Employers usually show their appreciation to employees with creative ability by offering them promotions and assigning duties resulting in increasing prestige and pay. Unfortunately, many employers fail to bring as many engineers into contact with research and new problems as they should for their own best interests.

General Procedure for Invention

To invent most effectively requires orderly thinking. Probably the best pattern to follow is to:

(*a*) Exercise one's powers of observation and try to arrive at an understanding.

(*b*) Train the memory to be selective with respect to observations and information within all fields of interest and possible usefulness.

(*c*) Analyze, coordinate, classify and associate knowledge acquired within the fields of interest.

(*d*) Having become interested in a particular problem, concentrate on it and keep working at it until a satisfactory solution is attained.

Every worth-while invention should be reduced to practice and thoroughly tested before it is placed in extensive use; and one should avoid the inclination to make changes too frequently after the product is put to use, even though they may appear to be improvements.

It has been noticed that people who have shown ability to develop ideas, to lead industry, and to invent, have had certain characteristics in common—they have been doers and workers, regardless of the extent of their formal education; they have had good memories and an accumulated mass of knowledge which always seems to be available to meet immediate problems.

Instinct is the spontaneous selection of a means to fulfill a need. Many competent people use the instinctive approach to problems in their everyday work.

Experiences Leading to Development of the Bailey Steam Flow—Air Flow Meter

I was fortunate in obtaining first-hand experience in the firing of slabs, sawdust, shavings, corncobs and coal in my father's sawmill in Ohio during the 1890's. Following this I gained valuable experience in firing about a hundred boiler tests, burning several different kinds of coal, at Ohio State University, 1900–1903. During the following seven years opportunity was provided to take part

* Past president of the American Society of Mechanical Engineers, founder of the Bailey Meter Company, and vice president of Babcock & Wilcox Company.

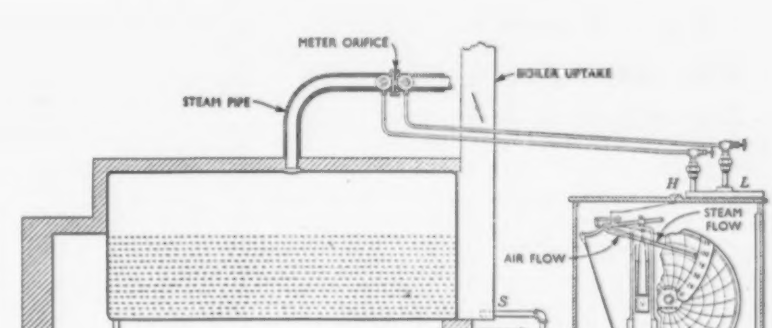
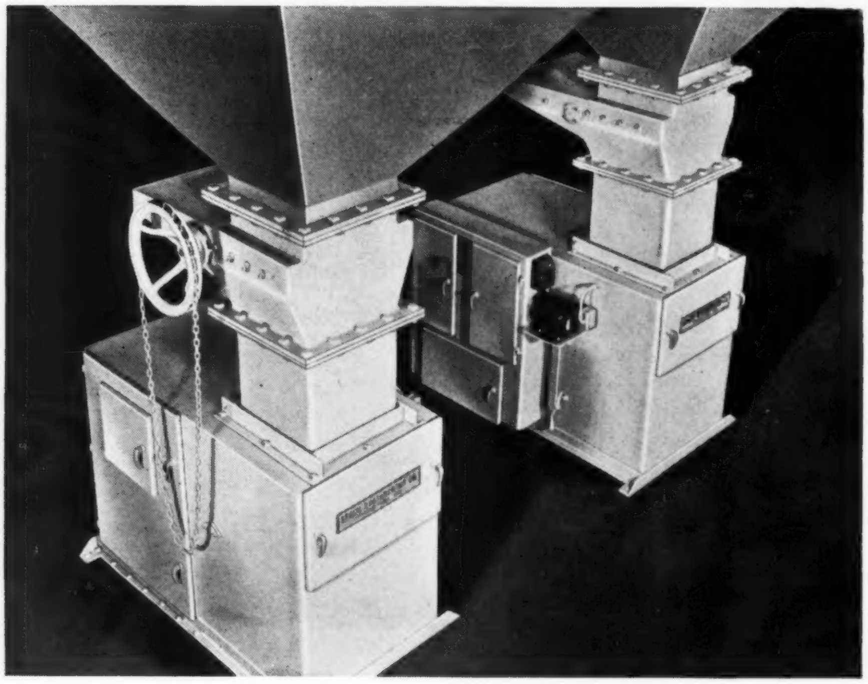
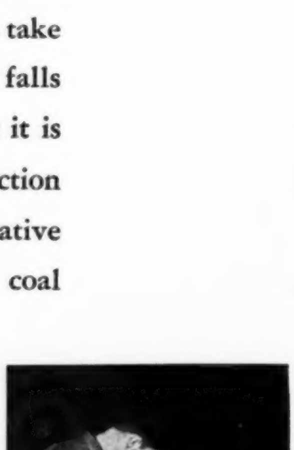


Fig. 1—Arrangement of boiler meter





in and observe the operation and testing of stationary, marine and locomotive boilers, as well as a variety of industrial furnaces. The fuel was mostly bituminous coal, hand-fired, with some stoker-fired boilers and pulverized coal in cement plants. Very few firemen had any instruments to guide them, but all were eager to learn about combustion efficiency through use of draft gages and the Orsat, which were carried as portable test equipment.

In order to meet the need for some permanent combustion guide, I started working on development of an automatic continuous gas analyzer; but before completing it, carbon-dioxide recorders became available. The early experience with these recorders left much to be desired, largely because knowledge of the one component, carbon dioxide, was inadequate for the needs of the fireman.

During firing of the boiler tests at Ohio State University it had been observed that the Ellison draft gage connected to the furnace responded to the condition of the fuel on the grate. This draft reading was not only responsive to the resistance to flow of air through the fuel bed, but also to the rate of output. Finally, after more than a year's study and further tests, the problem of using draft readings to operate a reliable combustion guide was on its way toward a solution. The resulting furnace indicator was based upon the principle of the Wheatstone bridge, used for measuring electrical resistances, but it employed the draft differentials to determine the fuel-bed condition by measuring the resistance of the bed to the flow of air. The grate bars used at that time had about 50 per cent air space and offered little resistance to air flow, compared with the fuel bed.

The furnace indicator, useful as it was for its original purpose of recording the fuel-bed condition, did not measure the quantity of steam produced. This, from the fireman's point of view, was far more important than to keep the recorder pen within the shaded band of the chart. It was desirable for the recorder to show the entire story of boiler operation; and, after many months of development and trial installations, a steam flow meter was developed for use with pitot tubes. Later the orifice was explored and developed; and subsequently use of a combined bell and displacer was adopted.

Differential Draft Loss Important

Steam flow, as a measure of output, was valuable but alone it gave no indication of combustion efficiency. Also, as furnaces became larger and the multiple-retort underfeed stoker became temporarily popular, the furnace indicator was less helpful, because the higher resistance of the grates, with their small air space, diminished the relative effect of the fuel bed itself. The furnace indicator was therefore doomed to be discarded, but the differential draft loss across the boiler passes had become recognized as a valuable indication, because it varied closely with the boiler output.

By that time it was recognized that a combustion guide could be made by comparing air-flow reading with steam flow, it being necessary to design the air-flow

mechanism so that it could be adjusted to read the same as the steam flow when the most economical combustion conditions existed. The combustion reaction may be calculated in terms of Btu per lb of air as well as in terms of the usual calorific value of the fuel.

The Bailey boiler meter which records steam flow and air flow on the same chart is represented by the sketch in Fig. 1 and in principle by Fig. 2. With the boiler meters currently in use, the air flow is adjusted so that the steam-flow and air-flow pens read the same ($f\sqrt{F - S}/\sqrt{H - L} = 1.0$) when the correct combustion conditions exist, regardless of the rate of steam output. The meter can be calibrated by making a series of combustion tests and adjusting the factor f of the air flow for each individual boiler and fuel.

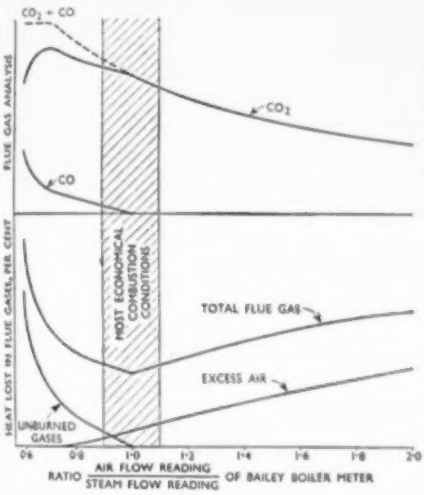


Fig. 2—Combustion characteristics illustrating principle of the boiler meter

Recently developed oxygen recorders are becoming recognized as valuable guides for all conditions where combustion of fuel takes place. They are often installed in addition to the steam flow-air flow or the fuel flow-air flow recorders as continuous checks, to aid in starting up and to meet unusual operating conditions.

It is difficult to see how the steam flow-air flow meter could have been invented if it had not been preceded by the furnace indicator.

After pulverized coal came into extended use, larger boilers were possible; means for supplying air and fuel to the furnace became more complicated; and the need for automatic combustion control became essential. However, the development of such control did not require invention of the same degree as was needed for the early steps in development of the boiler meter, because by that time the relationships between the different factors were known, and it was largely a problem of harnessing them to the instruments. There was need for many secondary inventions by many individuals in order to adapt the instruments through intermediate mechanism to the heavy motor or piston-operated dampers, without interfering

with the accuracy of the meter reading and chart records.

Development and use of the three-element feedwater control was deferred several years after the fundamental requirements were fully known, as it was desired first to verify the accuracy and reliability of the flow meter and level recorder type of mechanism.

Course in Air Pollution

In line with the widespread attention that is now being directed throughout the country to smoke abatement and air pollution, the School of Public Health at the University of Michigan has scheduled a short 3-day lecture and discussion course in air pollution for February 6, 7 and 8, 1950. In scope the course will deal with contaminants, factors involved in pollution and correction, physiological effects, air-cleaning devices, smoke and pollution control. Those selected to give the various lectures are all authorities on different phases of the subject, and following each group of talks there will be a panel discussion with a competent moderator to direct it along constructive lines.

While the course is laid out especially for the benefit of those charged with pollution control, health officers, industrial chemists, physicians of industrial communities, and sanitary engineers, others interested in the subject will be welcome. The enrollment fee will be $5.

Underground Gasification of Coal*

Gas produced by burning coal underground consists of a mixture of nitrogen, carbon dioxide, carbon monoxide, hydrogen, methane and some higher hydrocarbons. The nitrogen is in the incoming air and goes through essentially unchanged. The carbon dioxide results from complete combustion of carbon with oxygen, while the carbon monoxide is a product of the partial combustion of carbon with oxygen. The hydrogen is derived directly from the coal or from the decomposition of steam in contact with hot carbon monoxide or carbon. The methane is produced by synthesis during the gas-making reaction or together with higher hydrocarbon gases from distillation of the coal.

Heat is available from these gases in two forms—first, as sensible heat when they are cooled from a high temperature to a lower temperature; and second, as heat of combustion when the gases are burned with additional oxygen. The nitrogen and carbon dioxide can only give up sensible heat, while the carbon monoxide, hydrogen and methane can give up sensible heat and be burned as well.

For the generation of steam, with which to operate a steam turbine-electric generator at the site of the underground gasification, both sensible heat and heat of combustion could be used. The boiler plant should be close to the exit of the underground gasification operation, and the gases taken to it at as high a temperature as possible.

In utilizing the gaseous products of underground gasification in a gas turbine, the turbine can be located at the gas outlet, the sensible heat extracted through suitable heat-exchange equipment, and the heat of combustion obtained in the combustion chamber of the turbine. The overall efficiency of such a gas turbine installation may be 25 to 50 per cent greater than is now obtainable from present mining methods followed by coal utilization in a steam power plant.

To produce a synthesis gas for making oil and gasoline from underground gasification, it probably will be necessary to use oxygen instead of air underground. This would give a gas high in carbon monoxide and hydrogen concentration, and eliminate the inert nitrogen. It is contemplated that the resulting gases would be synthesized by the Fischer-Tropsch Process to give gasoline and diesel oil.

There are a number of other possible uses for the products from underground gasification; for example, in the production of chemicals or higher Btu gases, but

* Excerpts from a paper by Milton H. Fies of the Alabama Power Company presented on November 30, 1949, before the New York Society of Security Analysts.

no extended analysis has yet been made of these potentialities.

In the second experiment at Gorgas (Alabama) we have had the benefit of the judgment and advice of an eminent scientist, Dr. E. T. Wilkins, of the Department of Scientific and Industrial Research of the British Government. British engineers will undertake an experiment in underground gasification in the near future. Belgian engineers have their second experiment under way at Wandre, near Liege. French engineers are collaborating, and the Belgian coal industry is assisting the financing, together with the Belgian and French governments. In addition, French engineers are undertaking an experiment in Morocco at Djerada.

The two experiments at Gorgas have been in a flat seam. Most of the coal seams in the United States are relatively flat and, although it appears to be more difficult to gasify coal in the ground when lying in a flat bed, we have deliberately chosen these horizontal seams for experiments in order to find the process best adaptable to them. We have chosen the hard way.

The Russians state that large industries located near the Donetz Basin are obtaining their fuel supply from gas obtained by burning coal underground. We have no means of reliably checking these claims.

In the preliminary experiment in underground gasification at Gorgas, the following results were obtained:

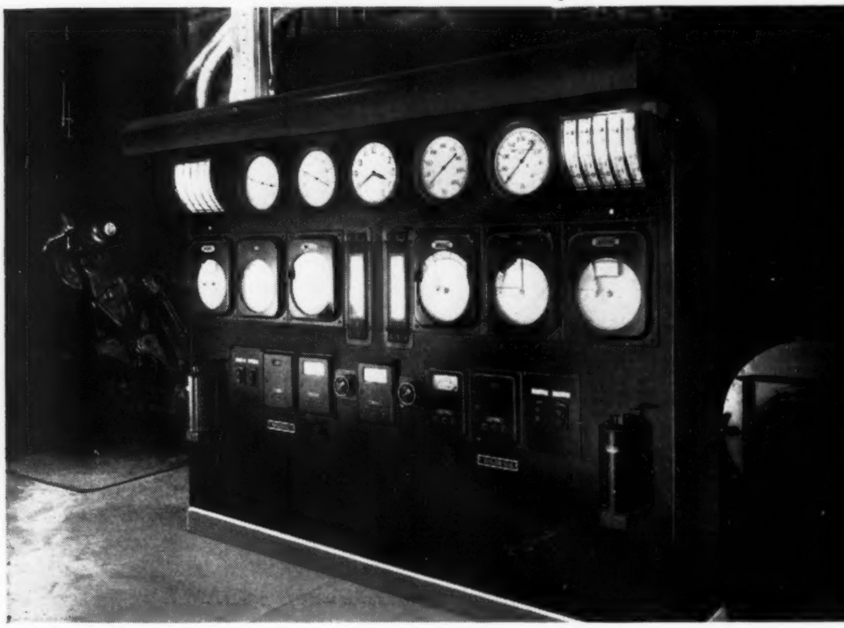
1. There was no difficulty in maintaining combustion of coal underground.

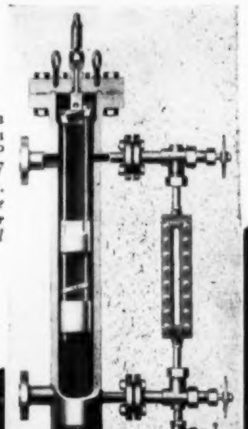
2. Coal in place was gasified completely. Examination of the underground residue showed that only ash and clinker remained in the combustion zone. No islands of unreacted coal or coke were found.

3. The high temperature developed by the gasification brought about changes in the overlying strata that appeared to be favorable to the process. Roof rock became plastic, expanded, and settled on the mine floor directly behind the reacting coke face. Settlement of the roof rock forced the air and gas to pass through the narrow openings along the coke-rock interface.

4. A gas of varying quality was produced by the several methods employed. In essence, it appears entirely possible that a power gas can be produced during combustion either by the use of air, air-steam, or air-enriched oxygen. A synthesis gas apparently can be made by using an oxygen-steam blast or possibly through a cyclic operation employing a steam run.

Accordingly, the second experiment was undertaken, beginning on March 18, 1949, and it has progressed. The Alabama Power Co. made available some 100 acres of coal land and technical assistance, both without cost to the government. The Bureau of Mines has planned, supervised and directed the work and financed the services and facilities used in the preparation and operation of the project. It has been estimated that the government alone will have expended, through June 30, 1950, some $750,000 in the experiment.





Tentative Program for World Power Conference

INFORMATION from abroad indicates substantial progress in arrangements for the Fourth World Power Conference which will be held in London during the week of July 10 to 15. This is the first Plenary Meeting of the Conference since that held in Washington in 1936, and its theme will be "World Energy Resources and the Production of Power."

Officers of the Conference are: Dr. W. F. Durand (U. S. A.), president; Sir Harold Hartley (Great Britain), chairman; Dr. Gano Dunn (U. S. A.), Monsieur Ernest Mercier (France) and Shri A. N. Khosla (India), vice chairmen; and C. H. Gray, secretary. H. C. Forbes, 4 Irving Place, New York 3, N. Y., is secretary-treasurer of the Executive Committee of the U. S. National Committee.

Papers will not be read at the technical sessions, although they will be printed and distributed to those in attendance in advance of the Conference. Thus, the technical sessions will be given over to the presentation of summaries by General Reporters whose further duty will be to promote discussion. Papers in full, general reports and summaries of discussions will be reproduced in the *Transactions*.

The two working languages of the Conference are English and French.

Topics to be covered by the technical sessions will be grouped in three divisions as follows:

Division I—Energy Resources and Power Developments, to consist of a single report from each of the National Committees.

Division II—Preparation of Fuels (solid, liquid, gaseous, by-product and waste).

Division III—Production of Power (steam raising and utilization, internal combustion engines, gas turbines, jet engines, hydro power, wind power, solar energy and atomic energy).

A total of 155 papers from 27 countries are scheduled for presentation. The complete list of titles and authors is not available at this writing but the following constitutes those scheduled for American authors:

Division I: Energy Resources and Power Developments

"Energy Resources of the United States," by W. E. Wrather, Jas. Boyd, Nelson Lee Smith and Eugene Ayres.

Division II: Preparation of Fuels

"Trends in Mechanical Mining and Preparation of Coal," by J. B. Morrow and Henry F. Hebley.

"Advances in Petroleum Refining," by W. M. Holaday.

"Synthetic Liquid Fuels" (to include shale oil), by W. C. Schroeder and A. C. Fieldner.

"Transportation, Storage and Peak-Load Capacity in the Natural Gas Industry," by George H. Smith and others.

"Production of Combustion and Synthesis Gas from Coal and Petroleum," by L. L. Newman, L. D. Schmidt and H. R. Batchelder.

Division III: Production of Power

"Design of Modern Steam Generating Units for High-Pressure, High-Temperature Service," by Wilbur H. Armacost.

"Trends in American Boiler Performance Requirements," by W. H. Rowand.

"Trends in Developments in Steam Turbine Practice for Central Station Service," by C. B. Campbell.

"Progress in the Design of Steam Turbines for Electric Power Generation in the United States," by Edwin E. Parker.

"Mercury Cycle Power Generation—A Progress Report," by H. N. Hackett.

"Operating Experience with High-Pressure, High-Temperature Steam Central Stations," by P. W. Thompson.

"Summary of Current Developments in Large Diesel and Gas Engines," by Ralph L. Boyer.

"Summary of Current Developments in Small Diesel Engines," by M. R. Bennett.

"American Gas Turbine Practice," by L. N. Rowley and B. G. A. Skrotski.

"Aircraft Gas Turbines," by J. S. Alford.

"State of Hydro-Electric Power Development in the United States," by E. Robert deLuccia and Frank Lloyd Weaver.

"Hydro-Electric Power—Available Power and Delivery of Water to the Power House," by William P. Creager, William F. Uhl and Byron O. McCoy.

"Hydro-Electric Power—Design and Operation of Power House and Equipment," by Earl B. Strowger.

"Heat Pump Progress in the United States," by Philip Sporn and others.

A number of tours in the United Kingdom, each lasting about a week, have been arranged for the week beginning Monday, July 17. Advantage has been taken in arranging the itineraries for the inspection of technical installations to include also places of historical interest and scenic beauty. For those not wishing to join one of these tours there will be full-day and half-day excursions from London. Numerous social functions are also scheduled.

The membership for those attending the Conference will be £6, this amount covering the cost of a set of preprints of the individual papers, copies of the General Reports, admission to all sessions and attendance at the official banquet as guest of the organizing committee. Membership fee for ladies accompanying members is also £6 and includes, besides admission to the opening and closing sessions and the banquet, visits to historical places in the neighborhood of London. The tours are variously priced.

Membership application forms and detailed information may be had from the office of the Secretary, 414 Cecil Chambers, 76/86 Strand, London W.C. 2, England. Travel and hotel accommodations may be obtained through Thomas Cook & Son Ltd. who have travel bureau offices in the principal American cities.

BOOKS

1—How to Keep Invention Records

By Harry A. Toulmin, Jr.

78 pages Price $2.50

The text discusses in easy readable nontechnical language the high spots of patent technique and particularly emphasizes to members of the bar, inventors and corporations the grave necessity of keeping adequate records of their valuable industrial property, known as inventions, the monopoly of which they are seeking for a period of years.

In the first part of the book the general nature of industrial property and monopolies granted to protect it are discussed, comprising such general fields as patents and copyrights. A discussion of foreign protection is also included.

In the second part a system is formulated based on the statutes as interpreted in numerous court decisions, to determine accurately the first user or inventor, and a practical method of insuring the recording of necessary dates is presented in a series of a dozen forms. These forms include Summary Card, Preliminary Sketch, Sheet, Research Record, Drawing Form, Construction Record Sheet, Test Record Sheet, Statement by Witness, Photographic Record Form, and others.

The final part deals with methods of patent investigation discussing briefly points such as Anticipation of an Invention, Purchase of Patents, Infringement, Validity of the Patent, Title, Investigation before Invention, etc.

2—Steam, Air and Gas Power

By W. H. Severns and H. E. Degler

509 pages Price $4.75

This is primarily a textbook for students in heat-power engineering courses, with a number of selected problems following each chapter. The treatment follows the usual sequence with introductory chapters on heat and heat power plants, followed by the principles of thermodynamics and discussions of fuels, steam generation, water conditioning, draft, prime movers, compressed air, internal-combustion engines, etc. Much of the text of the previous edition has been revised and some illustrations replaced. New material on the gas turbine has been added.

3—Combustion Engines

By Arthur P. Fraas

440 pages Price $5.50

This is a new text covering commercially important types of heat engines in which the working fluid consists of the products of combustion of hydrocarbons and air. While major attention is devoted to gasoline and diesel engines, some consideration is afforded the gas turbine.

The book is intended for the use of engineering college seniors and engineers in industry, and to satisfy such a diversity of interests an attempt has been made to relate practical considerations to basic theory. Toward this end photographs illustrating recent developments in engine testing techniques have been included, along with the extensive theoretical data. Numerous problems and pertinent technical references follow each chapter.

Following are some of the topics presented: engine types and construction, thermodynamics of engine cycles, fuel metering and injection, ignition, lubrication, cooling supercharging, performance analysis, overhaul and maintenance, gas turbines and engine installations.

4—Boiler Fireman's Handbook

By Joseph R. Darnell

200 pages Price $3.00

As the title implies, this is a practical treatment of the many problems encountered in boiler plant operation, much of the subject matter having been published serially in *Power Plant Engineering* during the period 1944–1946.

An idea of the coverage may be had from a listing of the chapters which include: Fundamentals of Combustion; Why Flue Gas Temperature Goes Up When the CO_2 Goes Down; Sampling and Analyzing Flue Gas; Interpreting Flue Gas Analyses; Measuring Flue Gas and Furnace Temperatures; Coal Storage and Preparation for Use; Boiler Efficiency Calculations from Flue Gas Analysis and Temperature; Types of Air Preheaters and Effect of Preheated Air, Natural and Mechanical Draft; Draft Gages; Hand Firing Methods; Stoker Firing; Oil Firing; Gas Firing; Pulverized Coal Firing; Waste Fuels; Heating Feedwater; and Flexibility in Firing Equipment.

5—Thermodynamics First Edition

By Edward F. Obert

571 pages Price $5.50

Many textbooks have been written on the subject of thermodynamics, but this one has been made more complete than the average with a view to extending the coverage beyond the usual undergraduate requirements, without, however, in any way sacrificing the fundamentals. The author, who is associate professor of mechanical engineering at Northwestern Technological Institute, Northwestern University, points out that this was done to encourage the student to progress beyond the borders prescribed by the instructor.

Many of the chapters have been developed as separate entities, of which that on "power cycles" is particularly good. These chapters deal with dimensions and units, fundamental concepts, the first law of thermodynamics, the reversible process, the second law of thermodynamics, properties of fluids, characteristics of gases, approximate calculations for real gases, flow of fluids, mixtures of gases and vapors, thermochemical calculations, power cycles (both vapor and gas) and refrigeration.

An extensive appendix contains numerous tables, curves and charts.

6—Elementary Steam Power Engineering

By Edgar MacNaughton

Third Edition

640 pages 6×9 Price $6.50

The general presentation resembles that of most textbooks, except that there is some departure from the usual sequence of topics by having the practical discussion precede the theoretical; and the text is more profusely illustrated. Each chapter is followed by a number of carefully selected problems.

Major changes since the previous 1933 edition include ASA symbols and abbreviations, abridgment of the latest steam tables, rewriting of much of the text to accord with advancing practice, and the inclusion of additional material, especially on thermodynamic principles and on turbines.

CE-S Establishes Western Division

A Western Division of Combustion Engineering-Superheater, Inc., New York, with headquarters in Los Angeles and branch offices in San Francisco and Seattle, was established, as of January first, as an expansion of the Company's facilities to meet the growing industrial activity of the West Coast states.

Robert M. Hatfield, Jr., formerly assistant general sales manager of the Company, has been appointed general manager of the Western Division and will be in charge of all activities in the states of Washington, Oregon, California, Nevada and Arizona. An engineering graduate of Purdue University, he has had 15 yr of service with the Company, interrupted during the war years when he was on loan to the War Production Board where he served successively as Chief of Boiler Section, Director of the Production Scheduling Division and Deputy Vice Chairman for Production, in charge of production for all war industries.

Robert M. Hatfield

H. G. Thielscher, formerly resident engineer of the West Coast, has been made chief engineer of the Western Division with headquarters in San Francisco. Prior to his association with Combustion, Mr. Thielscher was general superintendent in charge of power generating stations for the Potomac Electric Power Company of Washington, D. C.

Frank Bader of Combustion's Philadelphia Office has been transferred to San Francisco, and **Elmo Keeler** of the New York Office has been transferred to Los Angeles. **Hugh Nickle,** formerly in charge of the Company's Paper Mill Division in New York, and more recently district manager for the Northwest, will continue in that capacity with offices in Seattle.

The activities of the Western Division will include sales, installation and service of the Company's line of boilers and related equipment for the utility, industrial and marine fields, as well as pulverizing and drying equipment for industrial plants, chemical recovery units and bark burning boilers for the pulp and paper industry, and sewage sludge drying and incineration equipment for municipalities.

Fly Ash Use in Road Construction

There appears to have been some confusion on one point in reporting the discussion at the session on fly-ash uses during the recent A.S.M.E. Annual Meeting. On page 48 of our December issue it was stated that fly ash is being employed extensively in concrete road construction in New Jersey. This is not correct. It is being used successfully in mud-jacking concrete roads that have settled. The aggregate which is pumped in under pressure for this purpose, consists of 40 per cent fly ash, 60 per cent sand and 5 bags of cement to each cubic yard. The use of fly ash in pavement work in Detroit is understood to refer to bitumastic road construction.

Largest High-Pressure Turbine-Generator

During the past year a number of large turbine-generators were installed in utility power plants. The largest so far put in service to operate at high pressure and high temperature was a 1250-psig, 925-F, tandem-compound, double-flow, 1800-rpm, General Electric unit of 165,000-kw maximum capacity which was placed in service at the Fiske Station in Chicago. The 176,470-kva generator is hydrogen cooled and incorporates eight vertical coolers, four along each side. The one piece stator weighs 398,000 lb.

A. V. Adamson Retires

On January first, A. V. Adamson retired from Combustion Engineering-Superheater, Inc., after twenty-five years as manager of its New York District Sales Office. However, he will serve in a consultive capacity to the Company.

A native Virginian and an alumnus of Virginia Polytechnic Institute, Mr. Adamson acquired his early engineering experience with the United Gas Improvement Company of South Bend, Ind., and the Delaware & Hudson Railroad Company. He joined the Locomotive Pulverized Coal Company in 1914, working in collaboration with the late V. Z. Caracristi in the design of the first application of pulverized coal burning on a locomotive. This installation for the New York Central was followed by a number of similar applications for leading railroads. Four years later he became associated with the first pulverized coal installation in a central power station, namely, the Oneida Street Station of the Milwaukee Electric Railway & Light Company. With the general acceptance of pulverized coal following the Lakeside Station installation of the same company in 1920, he engaged in designing pulverized-coal-fired plants in many sections of the United States, and served as an advisory engineer for Combustion Engineering Company from 1920 until his appointment in 1924 as New York district sales manager.

A. V. Adamson

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